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RESEARCH AND DEVELOPMENT OF MATERIEL

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ENGINEERING DESIGN HANDBOOK

CARRIAGES AND MOUNTS SERIES

BOTTOM CARRIAGES

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HEADQUARTERS, U.S. ARMY MATERIEL COMMAND DECEMBER 1962

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AMCP 706-344, Bottom Carriages, forming part of the Carriages and Mounts Series of the Army Materiel Command Engineering Design Handbook, is published for the information and guidance of all concerned.

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PREFACE

This Handbook has been prepared as one of a series on Carriages and Mounts. It presents information on the fundamental operating principles and design of Bottom Carriages.

This Handbook was prepared under the direction of the Engineering Handbook Office, Duke University, under contract with the U. S. Army.

The text and illustrations were prepared by The Franklin Institute, under subcontract with Duke University. Technical assistance was rendered by the U. S. Army Weapons Command and Watertown Arsenal.

Since preparation of the text of this Handbook, responsibility for design and for all other functions pertaining to Army materiel, including preparation of this series of Handbooks, has been assumed by the Army Materiel Command. Any indicated responsibility of the Ordnance Corps in this regard should be understood as the responsibility of the Army Materiel Command.

Information on resulting changes in Handbook designation, together with a current list of Handbooks, are contained on the inside back cover.

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LIST OF SYMBOLS

A_h contact area of hub

A_r contact area of ring segment

A_t contact area of turntable

a length of distributed load from bolster; length of spoke

b length of distributed reaction on float; half the length of ring

c width of float

d length (span) of float; radius of hub

e distance from center of load to edge of float at 0

F_a primary recoil inertia force; vertical traverse bearing load

F_b vertical bearing clip load

F_c axial force on trail

F_{Ff} vertical front slide load of front bolster

F_{Fr} vertical rear slide load of front bolster

F_g propellant gas force; elevating gear load

F_h horizontal pintle load

F_{jr} rear jack load

F_{js} side jack load

F_j jack load

F_L lower horizontal pintle load

F_n normal force on trail

F_{Rf} front slide load of rear bolster

F_{Rr} rear slide load of rear bolster

F_{rx} ring load on each segment of ring and spoke turntable

LIST OF SYMBOLS (Cont'd)

F_{TC} inertia force of top carriage due to secondary acceleration

F_t inertia force of tipping parts due to secondary acceleration

F_{tr} tie rod load

F_U upper horizontal pintle load

F_V vertical pintle load

F_2 inertia force of secondary recoiling parts

I moment of inertia of cross section

K total resistance to recoil; total resistance to primary recoil

k_o foundation modulus

L length of trail

L_p center to center distance of trail pin lugs

M_{bx} bending moment at hub of ring and spoke turntable

M_p moment about diameter of turntable due to ground pressure

M_s moment at end of trail due to float and spade loads

M_t overturning moment of turntable

p_m maximum allowable ground pressure

p_1 minimum ground pressure on turntable

p_2 maximum ground pressure on turntable

R total resistance to secondary recoil; radius; resultant

R_a vertical reaction on axle

R_b lower reaction on trail pin

R_F vertical ground reaction on turntable

R'_F vertical load transmitted by top carriage

LIST OF SYMBOLS (Cont'd)

$\mu R'_F$	frictional force on front bolster
R_f	reaction on float
R_j	reaction on front jack
R_R	vertical ground reaction on float
R'_R	load on rear bolster
$\mu R'$	frictional force on rear bolster
$\mu R'_R$	frictional force on slides
R_r	secondary recoil rod pull
R_s	reaction on spade
R_t	upper reaction on trail pin
r	radius of turntable
S_f	factor of safety
T_r	rifling torque
u	minimum unit reaction of float
V_{bx}	vertical shear at hub of ring and spoke turntable
W_{BC}	weight of bottom carriage
W_F	weight of front bolster
W_R	weight of rear support of float
W_t	weight of tipping parts
W_{TC}	weight of top carriage
W_{tr}	weight of traversing parts
W_2	weight of secondary recoiling parts
w	maximum unit reaction of float
Z_t	equivalent section modulus of turntable

LIST OF SYMBOLS (Cont'd)

β traverse angle; contact angle; simplified expression in equations for beams on an elastic foundation

ϵ angular trail spread

θ angle of elevation; angular deflection of hub of turntable

θ_{ax} angular deflection at ring of ring and spoke turntable

θ_{bx} angular deflection at hub of ring and spoke

λ characteristic of the equations for beams on an elastic foundation

ϕ slope of terrain

ϕ_x location of spokes

ψ trail slope

CARRIAGES AND MOUNTS SERIES

BOTTOM CARRIAGES*

I. INTRODUCTION

A. PURPOSE

1. This is one of a series of handbooks on Carriages and Mounts, which deals specifically with the bottom carriage. It discusses the types of bottom carriage, its components and their functions. It considers the requirements which each must meet and presents design data and procedures. Figure 1 shows a typical bottom carriage structure of a single recoil weapon.

B. DEFINITION AND FUNCTION

2. The bottom carriage is that part of a weapon which supports the top carriage and provides the pivot for the traversing parts. It and its components, including wheels, trails, and outriggers, provide the structural foundation for the weapon. When emplaced and during firing, the bottom carriage transmits all carriage loads to the ground. During transit it may become the chassis. As part of a double recoil system, it is retractable and hence is relieved of this function. It houses the complement of that part of the traversing mechanism associated with the top carriage. The secondary recoil mechanism of double recoil systems is also part of the bottom carriage.

II. TYPES AND OPERATING CHARACTERISTICS

3. Although the definition is specific enough, the immediate structure supporting the top carriage is not always identified as the bottom carriage. Such terms as pedestal and platform apply equally well. The lines of demarcation grouping the various structures according to their respective categories are nebulous and cannot be sharply defined. But, as their functions are similar, all are included in the general class of bottom carriage. Not only the basic structure but all components such as trails, outriggers, spades, and floats are also considered. These components also include the frames of the limber and bogie.

A. BOTTOM CARRIAGE OF FIXED EMPLACEMENT WEAPONS

4. Basically there are two types of bottom carriage; one for fixed emplacements, the other for mobile units. The fixed emplacement type includes the base of disappearing and Barbette carriages, and pedestals serving in the same capacity. As stated in Reference 1,† fluid warfare has rendered this type obsolete and detailed discussions become unnecessary. However, it should be stated that the same design procedures apply to this type as apply to the bottom carriages of mobile artillery.

B. BOTTOM CARRIAGE OF MOBILE WEAPONS

5. The bottom carriages for mobile weapons range from very simple to quite complex structures. In some light artillery mounts, the forward part of the trails serve as the bottom carriage. Other mounts employ simple attachments that are little more than brackets attached to the axle. Still others have structures that serve as both chassis and bottom carriages. The more complex units which include firing platforms, pedestals and bottom carriages serve as supporting structures for the general range of weapons from light through heavy weapons. Examples of the various types follow.

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* Prepared by Martin Regina, Laboratories for Research and Development of The Franklin Institute.
† References are found at the end of this handbook.

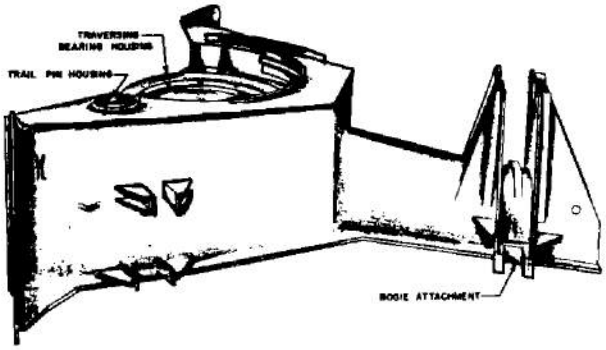


FIG 1. BOTTOM CARRIAGE, SINGLE RECOIL TYPE

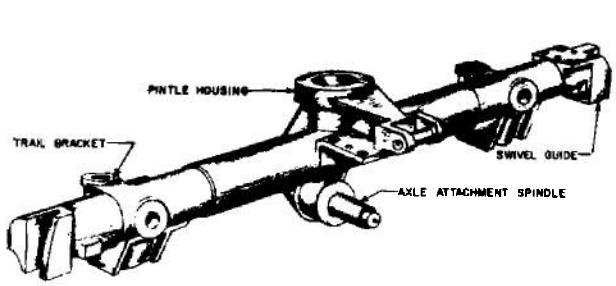


FIG 3. EQUALIZING SUPPORT

1. Supporting Bracket

6. The supporting bracket is a compact structure that serves as the bottom carriage of some light field pieces. It supports the top carriage and transmits its load to the ground via trails and axle supports. Figure 2 shows this type installation. Brackets on the axle provide attachment for trails, top carriage, wheels, and firing supports.

2. Equalizing Support

7. The equalizing support is a beam, usually tubular, with brackets for attaching top carriage, axle and trails (Fig. 3). The center bracket serves a twofold purpose by providing the pintle housing for the top carriage and the horizontal spindle about which the axle rotates to compensate for uneven terrain. The spindle transmits most of the vertical component of the firing load to the ground through the axle and wheels or firing support. The remainder of the vertical component and the entire horizontal component are transmitted by the trails which are attached to brackets located near the ends of the support. In addition to the three main brackets, two others, outside the trail brackets, provide vertical retaining grooves for confining axle rotation to one plane.

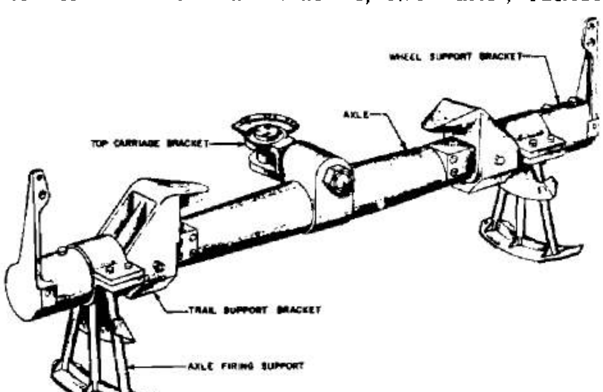


FIG 2. SUPPORTING BRACKET INSTALLATION

3. Firing Platform (240mm How., M1918M1 Type)

8. The firing platform, (Fig. 4) as the name implies, is a flat structure that supports the weapon during firing. It transmits all firing loads to the ground. Two of its components, the pintle and rear sector, transmit top carriage loads to the platform. The pintle functions as the pivot for the traversing parts and as a structural member supports horizontal and vertical loads. The sector has three functions. It supports vertical loads and provides the bearing surface for the rolling elements of the top carriage during traverse.

The main structure with three other components, namely, the trunk which fits into the emplacement pit and two outriggers with floats, is responsible for directing the firing loads to the ground; the trunk for the horizontal and the others for the vertical loads. The outriggers and floats, by effectively increasing the length of the base, lend stability to the weapon. The platform has a large recess immediately to the rear of the pintle. The recess formed by the hollow struc-

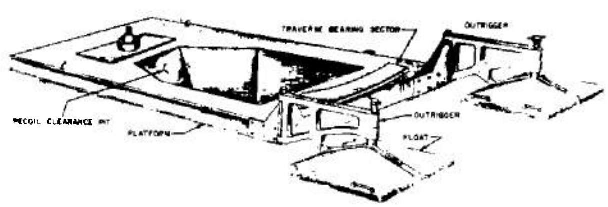


FIG 4. FIRING PLATFORM

ture of the trunk provides clearance for the recoiling parts. An emplacement of this type is limited to small adjustments in elevation and traverse.

4. Pedestal

9. The pedestal, Figures 5 and 6, more closely approaches the concept of a bottom carriage than any of the preceding supporting structures. It is the firing base and serves as the chassis during transport. This type is usually consigned to antiaircraft use as it is particularly suited for high angles of elevation in all directions. The structure is composed of three units. These are the leveling socket assembly, (Figure 6) the pedestal and four outriggers. The leveling socket is the immediate support of the top carriage and in turn is supported by the pedestal. Two retractable hydraulic jacks, fitted into outriggers, lower and raise the mount on and off the leveling socket.

10. The leveling socket is a circular structure and has the traversing gear, roller bearing, and clip ring assembled to its upper side. All are associated with the traversing parts. The gear is the fixed member of the traversing gear train. The bearing is the low friction element and the clip ring is the retainer that precludes tipping of the traversing parts. A hemispherical bearing is attached to the lower side of the leveling socket. It fits over its male counterpart on the pedestal and permits only tipping motion for leveling purposes. Leveling is accomplished by four leveling screws supported on special pads of the pedestal. The screws tilt the leveling socket and restore it to the horizontal plane (Figure 7). The angular displacements are small, on the order of five degrees, and can compensate for only minor irregularities of the terrain.

11. The pedestal conducts all forces to the ground, either directly or through its four outriggers. It becomes the chassis of the weapon during transport. The outriggers, being hinged to it, are used primarily during firing. Pins make the hinged joint rigid but for transport are withdrawn and the outriggers retracted

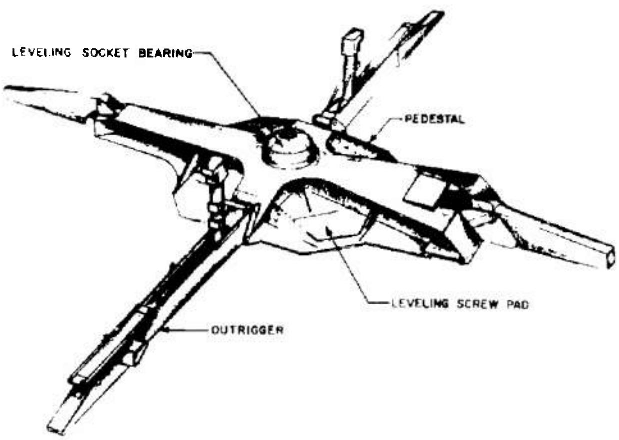


FIG 5 PEDESTAL

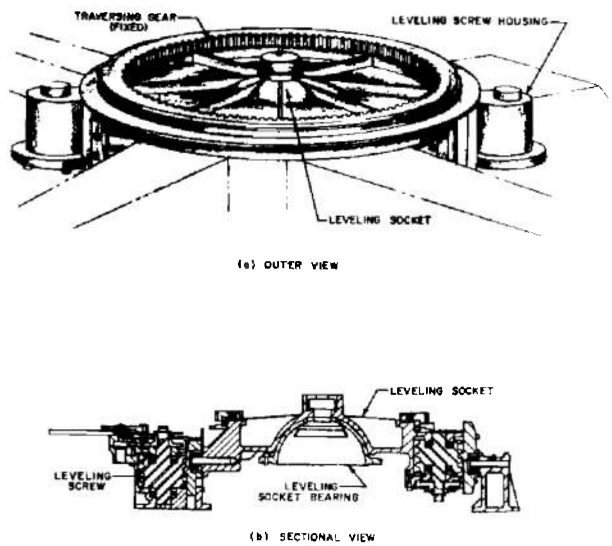


FIG 6 LEVELING SOCKET ASSEMBLY

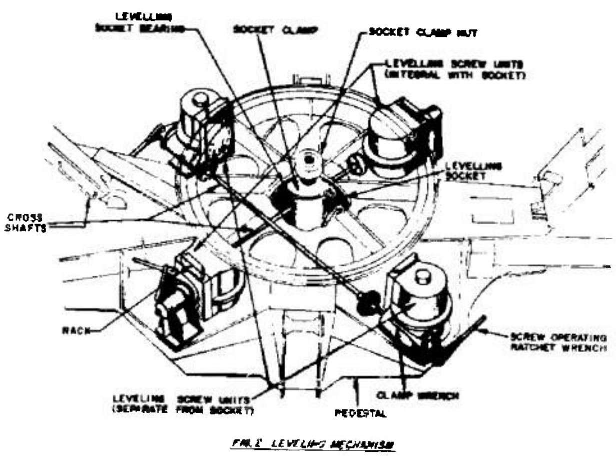


FIG 7 LEVELING MECHANISM

by folding them upright against the main structure. One outrigger becomes the prop for the gun tube while in this retracted position.

5. Bottom Carriage—Single Recoil

12. Bottom carriage may be a general term for all types of ground supporting structure or it may be a specific term such as bottom carriage for a single recoil weapon. In the single recoil weapon, the bottom carriage links the top carriage to the trails and forward supports during firing, and the top carriage to the trails and wheels during transport. No translation occurs between top and bottom carriages. Except for weapons equipped for unlimited traverse, split trails usually carry the recoil forces to the ground (Figure 8). Jacks or their equivalent bear the downward loads at the forward end. Only limited fine traverse is available after emplacement. It is necessary at emplacement to take care to orient the gun in the proper general direction.

13. Loads are transmitted from top to bottom carriage through the traversing bearing. Generally, radial bearings carry the horizontal components while thrust bearings carry the vertical components of the

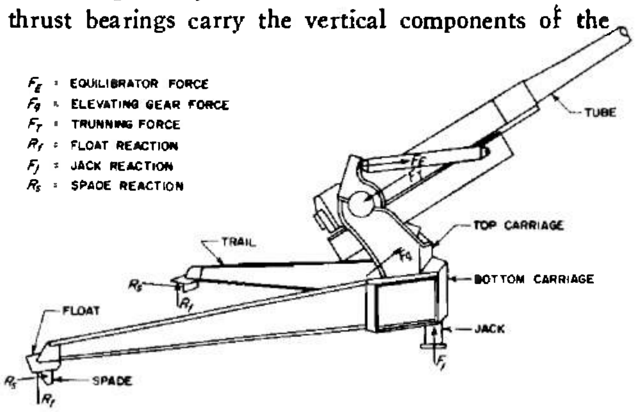


FIG 8 SPLIT TRAIL GUN SHOWING PRIMARY LOADS

firing loads. A discussion of various traverse bearings and their corresponding loadings analyses appears in another handbook.* An angular contact type bearing of large diameter and many rolling elements has proved satisfactory as a traversing bearing. It is a four point contact ball bearing. In addition to being capable of carrying axial and radial loads, it also is capable of transmitting bending moments such as the over-

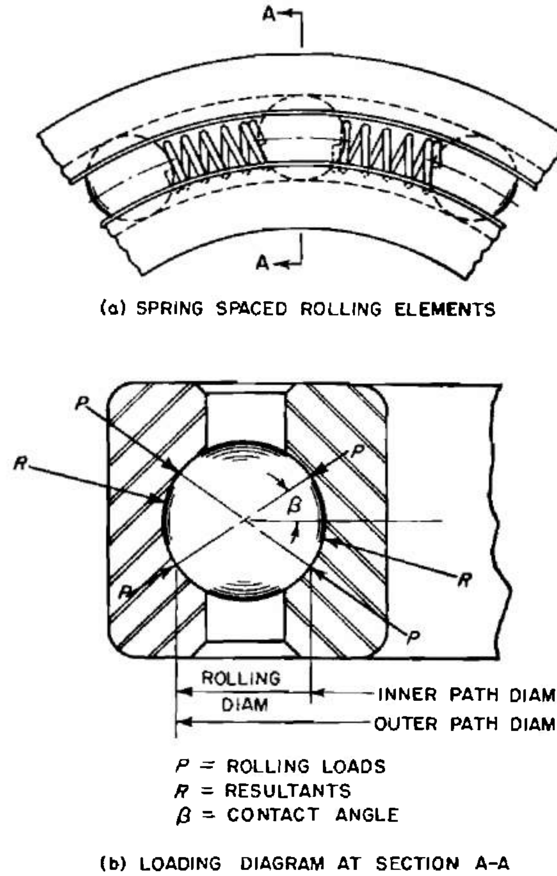


FIG. 9. FOUR POINT CONTACT BALL BEARING

turning moment during recoil. Figure 9 shows details of the rolling elements and the loading diagram. The spring separators serve to distribute torque more uniformly during traverse and provide for uniform support. Other advantages derived from the bearing are its ability to carry all firing loads whether or not uniformly distributed, its adaptability to either single or double recoil gun structures, and its low frictional properties.

6. Bottom Carriage—Double Recoil

14. The bottom carriage of a double recoil weapon (Figure 10) serves as the firing base conducting all forces to the ground, performs all traversing operations by having all essential equipment completely housed in its structure and houses the secondary recoil mechanism. Three basic units comprise its structure, the forward support called the turntable assembly, the rear support called the float assembly, and the tie

* Reference 2, Chapter IV.

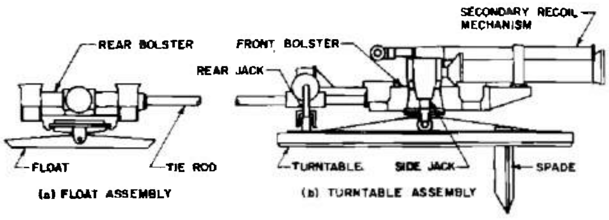


FIG. 10. BOTTOM CARRIAGE, DOUBLE RECOIL TYPE

rods which join the first two. Tie rods are slender members and therefore cannot lend any appreciable stiffness to the bottom carriage. Structural stiffness between turntable and float assemblies is provided by the side frames of the top carriage (Figure 15). Although the composite unit of top and bottom carriage is structurally rigid, longitudinal motion still occurs between the two during secondary recoil. Appropriate sliding surfaces where top carriage rests on bottom carriage provide for this motion. Clips in the same regions prevent lateral displacement. Slides and clips assure structural alignment between top and bottom carriage but are not involved in leveling assignments.

15. The bottom carriage functions as such only during emplacement. During transit it is suspended by the clips from the top carriage to be carried along with the other components of the weapon. It is rendered more readily transportable by shortening the distance between turntable and float assemblies, thus creating space for attaching the transporting units. This procedure is achieved by building the tie rods of telescoping members. The tie rod is attached, at each end, to the immediate supporting structure of the top carriage.

16. Each supporting unit, forward and rear, has a bolster which is the immediate support of the top carriage. The bolster in front is supported by the turntable while the one in the rear is supported by the float. Both turntable and float rest on the ground. To offset the effects of uneven or sloping terrain, bolsters are connected by what essentially are universal joints. The center of the turntable is the traversing axis. Coarse traverse is unlimited and is achieved by jacking the float off the ground and pushing the weapon around to the desired position. Fine traverse is limited and is achieved through the traversing mechanism assembled to the rear support.

17. The turntable is secured by spades and transmits part of the vertical and nearly all of the horizontal forces to the ground. Its vertical component depends solely on the geometry of the applied forces which include the weight of the weapon. The horizontal component comprises the rod pull of the secondary recoil mechanism, the frictional force of the forward slides, and the tie rod loads. The float transmits the remaining vertical forces. The horizontal force on the rear support comprises the frictional forces on the rear slides. However, the rear float is not staked and is capable of transmitting only that force which can be sustained by the frictional resistance between float and ground. The residual horizontal force is carried by the tie rods to the front bolster structure.

18. The bolsters, made of built-up beams, join top carriage to turntable and float and provide attachments for all units which transmit loads between these structures. The load transmitting units include the slides and clips, the secondary recoil mechanism, the rear and side jacks, the traverse bearing in front, and the structure containing the traversing mechanism in the rear. All jacks are attached to the front bolster and rest on a circular track near the edge of the turntable. Wheels on their lower ends roll on the track during traverse. The rear jack is located on the longitudinal center line of the carriage (see Figure 16) and performs the sole function of tilting the top carriage forward about the traverse bearing thereby lifting the float to provide the ground clearance needed for coarse traversing. The side jacks compensate for gentle slopes by leveling the bolsters and hence the top carriage, and provide lateral stability for the emplaced weapon when they help transmit the rifling torque (see Figure 16). The bottom carriage of a double recoil type structure is unique among bottom carriages in that it completely houses the traversing mechanism, including the traversing bearing.* The traverse bearing unit, at the center of the turntable, must be equivalent to a ball and socket joint to provide rotation not

* Reference 3, Chapter II, Part D.

only in the horizontal plane but also in the vertical planes. It must be capable of supporting both horizontal and vertical loads. Figure 11 shows two traversing bearings. One is a simple ball joint, the other has a ball joint for tipping motion, and a roller bearing for traversing.

19. The rear bolster and float are linked to each other by the traversing gear structure (see Figure 10(a)). This structure includes the gear box and the slide through which all loads pass to the float.* The upper member of the slide is integral with the gear box and the lower member is hinged to the float. A transverse shaft, connecting traversing structure to bolster and a longitudinal shaft connecting traversing structure to float, form a universal joint which permits the float to rest evenly on the ground. The mating members of the slide are sections of a circular annulus whose center is at the turntable axis. Fine but limited traverse is achieved through a gear train which terminates at the rack. This rack, parallel to the slide, is rigidly fixed to the lower member. As the gear train functions, the pinion moves along the rack to traverse the weapon.†

* **Reference 3, Figures 9 and 10.**

† **Ibid., Figure 9.**

III. DESIGN PROCEDURES AND OBJECTIVES

A. STRUCTURE

20. As for all structures, the bottom carriage design should be directed toward simplicity, symmetry, and compactness. These criteria are usually compatible, the realization of one stimulating the incorporation of the other two. All tend to keep the weight down; compactness through size alone. A compact bottom carriage also helps provide a low silhouette for the weapon. Symmetry means uniformly applied loads and ultimately a stronger, lighter unit since eccentric loads require heavy local reinforcements. If both applied loads and structure are symmetrical, deflections will be symmetrical with less adverse effect upon the weapon's accuracy than would otherwise occur if symmetry did not prevail. Where costs are involved, simplicity becomes a major factor. A simple structure means ease of manufacture, ease of assembly, and ease of maintenance, all conducive to low costs.

21. Another problem confronting the designer deals with the selection of material. The bottom carriage is transported, hence, lightweight structures are preferred. Highly stressed components should be made of material with high strength-weight ratios. If applied loads are relatively small but if the sizes of the components are inherently large, lightweight, low strength materials should be given preference.

Regardless of the choice, care must be exercised in selecting a material that is readily available. In case a shortage of the original choice is anticipated, it is sometimes feasible to design a structure on the basis of one material so that it can be easily modified to conform to the physical properties of another. However, this practice can be dangerous if, in trying to include too many variables, the net result becomes an inferior structure.

22. A prime asset of any weapon is a short emplacement time. Emplacement includes two general operations, the preparation of the terrain and the positioning of the weapon. Outside of selecting a suitable location which includes the condition of the ground, little can be done here to speed emplacement. Assuming that the terrain is favorable, any disturbance to it in order to position and secure the bottom carriage and its components will add to the emplacement time. This includes digging clearance holes for the recoiling parts

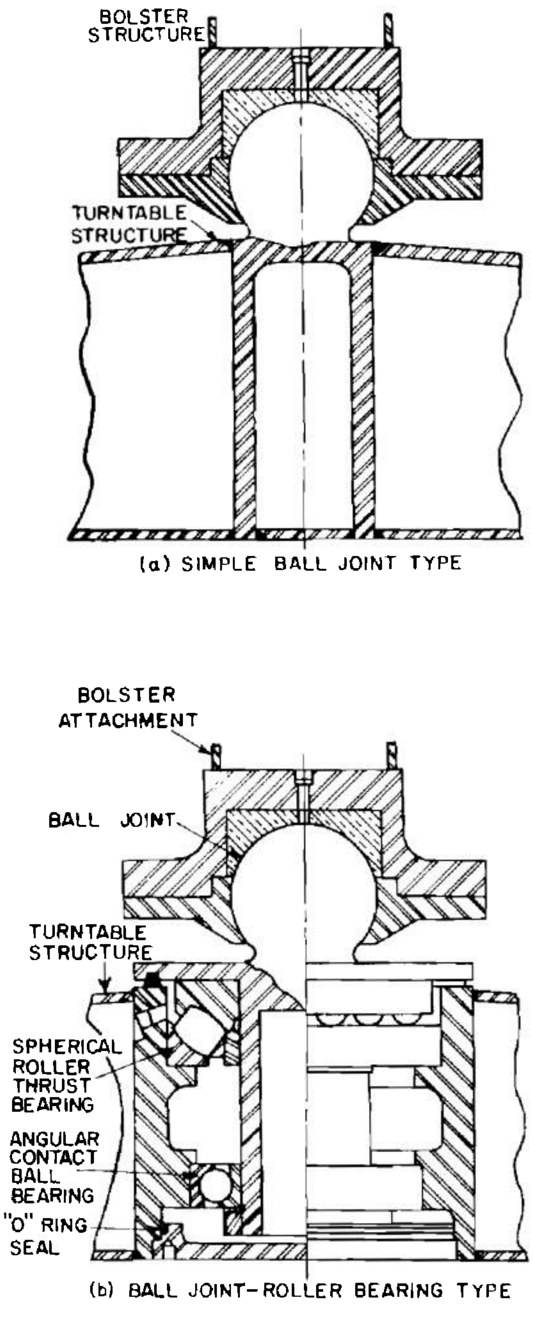


FIG. 11. TURNTABLE TRAVERSE BEARINGS

and holes for trail spades. A bottom carriage similar to those on double recoil weapons eliminates excavations of these types. Arranging the bottom carriage components in firing position sometimes requires considerable time, particularly for heavy structures thus creating the need for power operated equipment. Power from the prime mover is readily available or auxiliary units, either detached or assembled to the weapon, may be used. If the power units are assembled to the weapon, emplacement time will be cut further, at least by the duration of the hook-up period. Not only does powered handling equipment decrease emplacement time, but also reduces the number required for the operating crew to a minimum.

B. CONSTRUCTION—GENERAL

23. The type of construction of the individual parts whether forgings, castings or mill stock, should be left to the discretion of the designer. However, he should be guided by the relative suitability of each type for each application. Lightweight materials should be duly considered. Forgings demonstrate their greatest advantage where high strength-to-weight ratios are needed. High cost of forgings is the chief disadvantage. Castings may be used if weight is not critical. Large fillets moderate stress concentrations at re-entrant angles. After machining operations are completed, castings are usually ready for installation. It is assumed here that they have been stress relieved before machining to insure dimensional stability. Bulkiness and a lengthy manufacturing process are disadvantageous. Mill stock is generally used for built-up structures. Availability and low cost are its greatest assets. Weldments are preferred. They are relatively simple and light. Construction requires only a short time. Although weldments are prone to warp, this tendency is overcome by stress relieving through heat treatment. Regardless of the type, the method of construction and material selected must show compatibility with low cost, low weight, ease of manufacture, and availability of material.

C. CONSTRUCTION—SINGLE RECOIL TYPE

24. The bottom carriage or pedestal of a single recoil weapon is one compact structure, usually a casting or a weldment. The simpler types, namely the supporting bracket and equalizing support, may be added to this group. Provision must be made for the attachments of the top carriage and its associated parts and for the various ground supports and attachments required during firing and in transport. The former includes the pintle or its equivalent, some components of the traverse bearing, and either traversing gear or mechanism. The ground supports and attachments may include trails or outriggers, tow bar unit, jacks, and the transport units such as bogie and limber.

The pintle is the stanchion about which the top carriage rotates. Paired with the traversing bearing, it links the top and bottom carriages and is capable of transmitting both horizontal and vertical loads. Pintles are simple structures, little more than cylinderical pins, and are readily machined of forged steel.

25. Some weapons, particularly antiaircraft ones which are mounted on pedestals, substitute the more elaborate leveling socket for the pintle. The body of the socket (Figure 6) is a large, circular, concave, steel casting whose center is formed into a hemispherical socket. Its counterpart on the pedestal is also a steel casting (Figure 5). The two bearing surfaces are held snug by a steel clamp which extends through the mating hemisphere and is drawn tight by a nut bearing against the outer surface of the upper member.

26. Several types of construction are suitable for ground supports of bottom carriages. Trails and outriggers are built-up box beams. Either the complete structure may be made of mill stock, or the hinge at one end and the spade and float at the other may be forged or cast. Weldments are used in preference to riveted structures. Tow bar and steering units are usually of welded tubular construction. The moving parts of jacks are forgings or are machined from mill stock. Jack housings and axle firing supports are steel castings. The structures of bogies and limbers are well suited to be fabricated as weldments.

D. CONSTRUCTION—DOUBLE RECOIL TYPE

27. The bottom carriage of a double recoil weapon is ideally suited for weldments. It is largely composed

of built-up units, the individual structural members being readily cut from mill stock. Included in this group are the bolsters, turntable, float, and tie rods. The fittings joining these components and those attaching top to bottom carriage may be cast or forged or they may be small weldments. These fittings are the tie rod attachments, the traverse bearings, the slides and clips for the top carriage, and the secondary recoil attachment. Some may be welded to the larger structures but others must be bolted to them to facilitate assembly and maintenance.

IV. LOAD AND STRESS ANALYSIS

A. LOADING CONDITIONS

28. The bottom carriage is subjected to two types of loading condition, namely, firing and transporting. The major loads appear during firing and are transmitted by the top carriage. This does not eliminate transport loads from consideration but it does relegate them to secondary importance. All firing conditions should be investigated during recoil and counterrecoil and for all angles of elevation. Usually the maximum, minimum, and some intermediate angle provide the critical design loads. Transport conditions include normal travel, braking, and 30 percent side slope. The critical load is found by multiplying the actual load by the load factor; 1.5 for firing, 3.0 for sprung transport loads, and 5.0 for unsprung transport loads. The side slope condition is not prevalent and then only at low speeds, hence a load factor of 1.0 is adequate. Axiomatically, when stresses are computed for the actual applied loads, the load factor becomes the factor of safety. Transport conditions seldom yield critical design loads for the bottom carriage but they are the sole basis of design for the associated structures of bogie, limber, and towing unit.

B. SINGLE RECOIL TYPE—FIRING CONDITION

1. Loading Analysis—Split Trail

29. Diagrams of loading conditions are shown in Figure 12 for a gun carriage whose top carriage is attached to an equalizing support (Figure 3) and in Figure 13 for a gun carriage whose top carriage is supported by a bottom carriage (Figure 8). The loading diagrams are essentially the same except for the applied loads. The overturning moment is resolved into a system of horizontal forces in Figure 12 and into one of vertical forces in Figure 13. The applied loads on the bottom carriage are equivalent to the reactions for the loading analysis of top carriages.*

* Reference 2.

2. Definition of Symbols

a = distance, center of pressure of spade to ℄ trail.

b = distance, center of pressure of float to ℄ trail.

c = distance, center of coordinate system to ℄ trail.

d = distance, center of coordinate system to horizontal plane of F_h or $F_{U'}$

e = distance, center of coordinate system to horizontal plane of $F_{L'}$ (Figure 12 only)

h = distance, ground level to center of coordinate system

x_a = distance, ℄ pintle to $F_{a'}$ (Figure 13 only)

x_b = distance, ℄ pintle to $F_{b'}$ (Figure 13 only)

y_a = distance, ℄ pintle to center of axle or jack

y_t = distance, ℄ trail pin to x-z plane (Figure 13 only)

F_a = rear load on traverse bearing from top carriage (Figure 13 only)

F_b = front load on traverse bearing from top carriage (Figure 13 only)

F_h = horizontal load on pintle from top carriage (Figure 13 only)

F_L = lower horiontal load on pintle from top carriage (Figure 12 only)

F_U = upper horizontal load on pintle from top carriage (Figure 12 only)

F_V = vertical load from top carriage (Figure 12 only)

L = length of trail

R_a = vertical reaction on axle

R_f = reaction on float *

R_j = reaction on front jack

R_s = reaction on spade *

β = traverse angle

ϵ = angular trail spread

ψ = trail slope

30. Since the load analyses are similar, only that for Figure 12 will be used. The reactions on the spades are the only resistance to the horizontally applied loads on the traverse pintle. Balancing the horizontal force components parallel to the y-z plane yields the expression

$$R_{s1} \cos\epsilon + R_{s2} \cos\epsilon = (F_U - F_L) \cos\beta \qquad (1)$$

* Note that in the analysis 1 and 2 identify the symbols with respect to the left and right trail respectively.

Similarly, balancing the horizontal components parallel to the x-z plane shows that

$$R_{s1} \sin\epsilon = R_{s2} \sin\epsilon + (F_U - F_L) \sin\beta \qquad (2)$$

Solving for R_{s2} in Equation 2

$$R_{s2} = R_{s1} - (F_U - F_L) \frac{\sin \beta}{\sin \epsilon} \qquad (3)$$

Substituting for R_{s2} in Equation 1 and solving for R_{s1}

$$R_{s1} = (F_U - F_L) \frac{\cos \beta + \sin \beta \quad \cot \epsilon}{2 \cos \epsilon} \qquad (4)$$

The float reactions and the reaction on the front support, in this case, the reaction on the axle, R_a, are determined by balancing the moments. The distribution of vertical moments follows the same procedure as that for the distribution of horizontal loads. Moments are taken about the intersection of R_a and the y-axis. Balancing the moments in planes parallel to the y-z plane gives the equation

$$\begin{aligned}(R_{f1} + R_{f2}) \; [(b + L \cos \psi) \cos \epsilon + y_a] = \\ (dF_U + eF_L) \cos \beta + y_a F_V \\ + (R_{s1} R_{s2}) (a + h) \cos \epsilon \qquad (5)\end{aligned}$$

In the planes parallel to the x-z plane, the moment equation becomes

$$\begin{aligned}(R_{f1} - R_{f2}) \; [(b + L \cos \psi) \sin \epsilon + c] = \\ : (dF_U + eF_L) \sin \beta \\ + (R_{s1} - R_{s2}) (a + h) \sin \epsilon \qquad (6)\end{aligned}$$

The front reaction is obtained by the summation of the vertical forces $R_a = F_V - R_{f1} - R_{f2}$ (7)

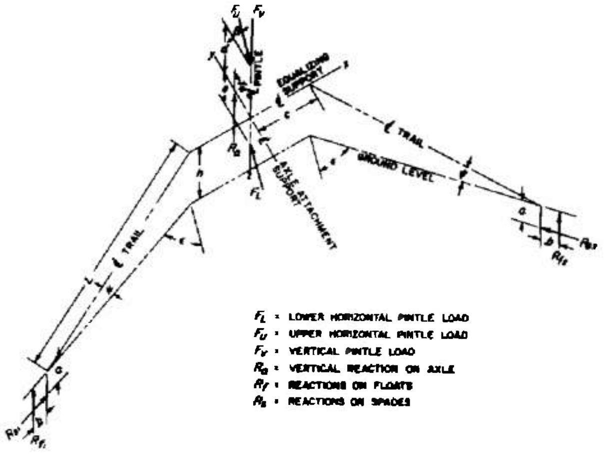


FIG. 12 LOADING DIAGRAM OF EQUALIZING SUPPORT-SPLIT TRAIL STRUCTURE

These equations are too complex to show the general solution for R_{f1} and R_{f2}. Much more convenient solutions are available by inserting the numerical dimensions. Maximum trail loads develop at the maximum traverse angle. The angle, ϵ, (Figure 12), should never be less than the traverse angle, β (Figure 12), indicating the direction of the resultant of overturning moment and rifling torque, otherwise lateral stability during firing cannot be assured.

31. The isolated trail with its loads is shown in Figure 14. The spade and float reactions are resolved into axial and normal loads and moment at the ground end, and into pin loads at the bottom carriage or equalizing support end. The axial or columnar load

is $F_c = R_f \sin \psi + R_s \cos \psi$ (8)

where R_f and R_s are the larger of the two float and spade loads of Equations 3, 4, 5 and 6.

The normal load is $F_n = R_f \cos \psi = R_s \sin \psi$ (9)

The induced moment is $M_s = bR_f - aR_s$ (10)

Pin reactions are determined by balancing the loading system. The upper pin reaction becomes

$$R_t = \frac{M_s + LF_n}{L_p} + \tfrac{1}{2} R_s \qquad (11)$$

The lower reaction is $R_b = R_t - R_s$ (12)

The reactions at the trail pins completes the loading system on the bottom carriage or its equivalent structure. Stresses for any critical region may now be computed to determine required structural sizes.

C. DOUBLE RECOIL TYPE*—FIRING CONDITION

1. Location of Bottom Carriage

32. Before a loading analysis can be made for any firing condition, the bottom carriage of the double recoil type must have the length of the firing base established and must be located with respect to the top carriage so that stability is assured during the complete recoil cycle. Stability is achieved only if neither base leaves the ground, specifically, the turntable during recoil or the float during counterrecoil. The activity during counterrecoil involves only small loads except for buffing when the loads may become critical. A schematic of the mechanics appears in Figure 15. All data are shown in the positive sense. The definitions of forces and reactions follow.

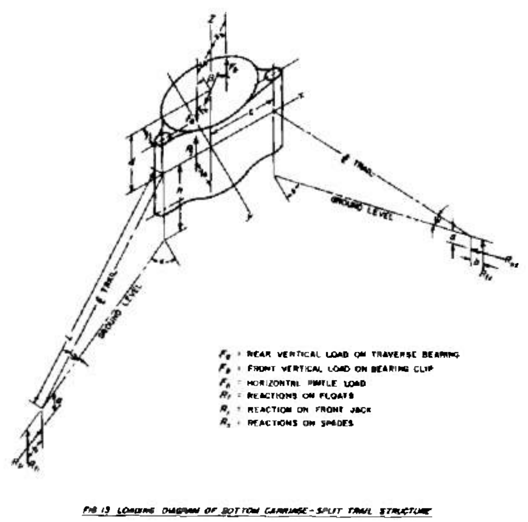


FIG. 13 LOADING DIAGRAM OF BOTTOM CARRIAGE-SPLIT TRAIL STRUCTURE

* The subject of double recoil is discussed in detail in Reference 4.

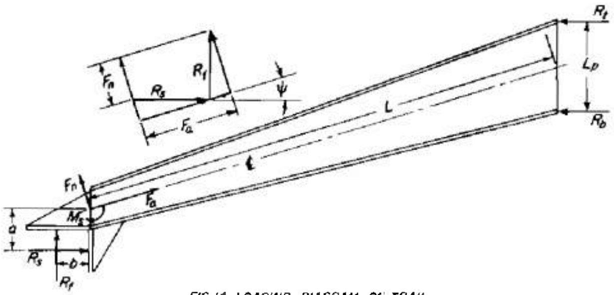


FIG 14 LOADING DIAGRAM OF TRAIL

F_a = inertia force of primary recoiling parts due to primary acceleration

F_g = propellant gas force

F_t = inertia force of tipping parts due to secondary acceleration

F_{TC} = inertia force of top carriage due to secondary acceleration

R_F = vertical ground reaction on turntable

R_R = vertical ground reaction on float

R_s = horizontal ground reaction on spade

W_{BC} = weight of bottom carriage

W_t = weight of tipping parts

W_{TC} = weight of top carriage

θ = angle of elevation

When adapted to a single recoil system all forces due to secondary recoil are zero but the analysis of the structure will not change.

33. The forces in Figure 15 are shown for the recoil accelerating periods. In other periods, the directions may be reversed but not necessarily simultaneously. The dynamics of the system are such that all forces are not always present. For instance, propellant gases are no longer effective when secondary recoil starts, therefore, F_t and F_{TC} appear after F_g becomes zero. However, the primary recoil force is always in the direction of F_g and is still very much in evidence as the secondary recoil forces appear, thereby inducing the largest overturning moment during recoil. This moment is not critical at high angles of elevation but become more so as the tube approaches the horizontal. The stability during recoil is checked by taking moments about the intersection of ground line with the center of the float.

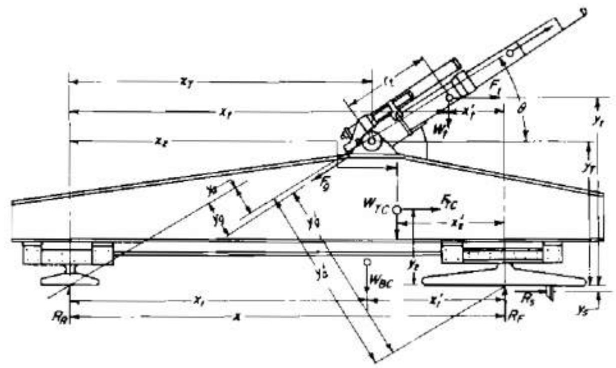


FIG 15 STABILITY STUDY DIAGRAM OF DOUBLE RECOIL WEAPON

$$xR_F = y_gF_g - y_aF_a + x_1W_{BC} + x_2W_{TC} + y_2F_{TC} + x_tW_t + y_tF_t - y_sR_s \tag{13}$$

Remember that F_t and F_{TC} are zero when F_g is still present.

Stability during recoil is assured when $xR_F \overline{>} 0$. At low angles of elevation, stability can still be maintained if the float is moved farther rearward to lengthen the firing base. But practical considerations preclude excessively long firing bases thus limiting the minimum angle of elevation.

To check the stability during buffing, moments are taken about the intersection of the ground line and the center of the turntable.

$$xR_R = y'_g F_g - y'_a F_a + x'_1 W_{BC} + x'_t W_t$$
$$- y_t F_t + x'_2 W_{TC} - y_2 F_{TC} + y_s R_s \qquad (14)$$

Only if $xR_R < 0$ will the weapon be unstable. Fortunately, the nature of the dynamics of the system are such that the primary recoiling parts are back in battery before secondary buffing starts, consequently, the two buffing forces never combine to nose over the weapon. Nevertheless, both primary and secondary buffing conditions must be investigated to learn which is critical. Moving the turntable sufficiently ahead of the CG is a simple matter to correct any nosing-over tendencies but a corrective measure of this sort not only increases the burden of the rear jack but also may lengthen the bottom carriage; two provisions that should be avoided.

34. Jack Loads. The rear jack raises the float off the ground by tilting the top carriage forward. Its location, on the longitudinal axis and resting on the outer rim of the turntable, must be to the rear of the center of gravity of the traversing parts. As the distance by which the jack is separated from the center of gravity becomes larger, the jack load becomes correspondingly smaller thereby reducing the burden on the jack and its supports and also the effort required for coarse traverse. Since the jack distance is limited by the size of the turntable, the center of gravity of the traversing parts should be located as near to the turntable axis as stability requirements during buffing permit. The rear jack load (Figure 16) is

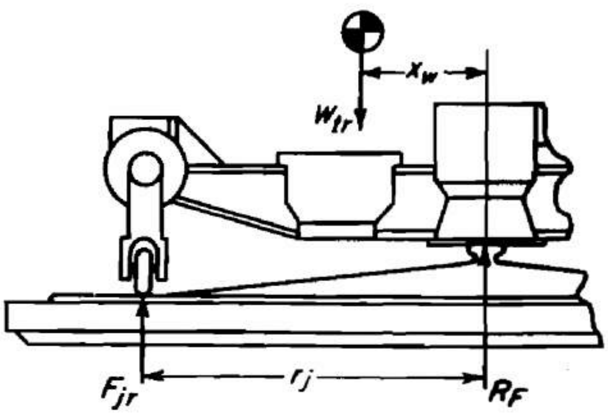


FIG. 16. LOADING DIAGRAM OF REAR JACK

$$F_{jr} = \frac{x_w}{r_j} W_{tr} \qquad (15)$$

where W_{tr} = weight of traversing parts.

Of the two side jack functions, only one sustains an appreciable load; the load induced by the rifling torque. From the geometry shown in Figure 17(a), the load on the side jack, assuming zero angle of elevation is

$$F_{js} = \frac{T_r}{r_j} \qquad (16)$$

where T_r = rifling torque

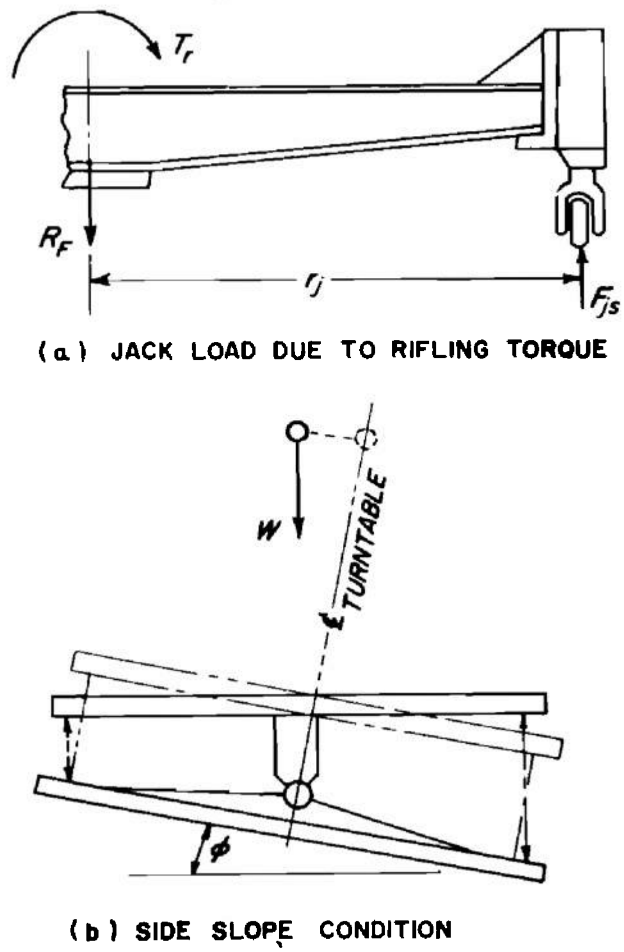


FIG. 17. LOADING DIAGRAM OF SIDE JACK

A much lighter jack load will level the weapon as it rests sideways on a gradual slope, ϕ. During the leveling process this load reduces to zero as the center of gravity moves toward the center of the turntable (see Figure 17(b)). When the center of gravity of the weapon arrives directly over the turntable, the side jacks are relieved completely of all static loads.

2. Bolster Loads

35. After the stability of the weapon has been achieved, the top and bottom carriages are treated as two distinct units and, except for their mutual attachments, are designed as such. The design loads on the bottom carriage appear as reactions in the top carriage load analysis*. The bottom carriage as a structure may be analyzed as two separate parts because neither front nor rear support has any appreciable influence on the strength of the other.

a. Rear Bolster

36. The vertical loads on the rear bolster are transmitted to the ground through the float. For the structural analysis, the entire frictional load on the rear support is conservatively assumed to be transmitted by the tie rods to the front support. Figure 18 shows the loading diagram. Taking moments about the intersection of F_{Rr} and $\mu\,R'_R$

$$x\,F_{Rf} = x'\,R_R - aW_R + yF_{tr} \qquad (17)$$

and solving to F_{Rf}

$$F_{Rf} = \frac{1}{x}\,(x'\,R_R - aW_R + yF_{tr}) \qquad (17a)$$

where

$F_{tr} = \mu R'_R$, tie rod load

$R_R = R'_R + W_R$, reaction on float

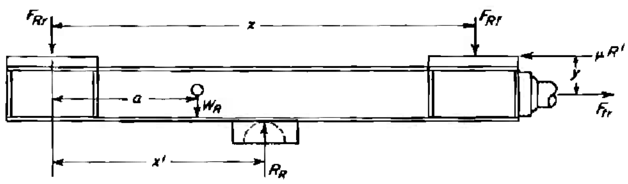
$R'_R = F_{Rr} + F_{Rf}$ = load on rear bolster

$\mu R'_R$ = frictional force on slides

W_R = weight of rear support

The vertical load on the rear slide

$$F_R = R_R - W_R - F_{Rf} \qquad (18)$$



F_{Rf} = VERTICAL LOAD ON FRONT SLIDE
F_{Rr} = VERTICAL LOAD ON REAR SLIDE
F_{tr} = TIE ROD FORCE
R_R = FLOAT BEARING VERTICAL REACTION
$\mu R'$ = FRICTIONAL FORCE ON REAR BOLSTER
W_R = WEIGHT OF REAR SUPPORT

FIG. 18. LOADING DIAGRAM OF REAR BOLSTER

b. Front Bolster

37. The loads on the front bolster are transmitted to the ground through the traversing bearing and turntable.

Figure 19 shows the loading diagram. All loads are known except the front and rear slide loads, F_{Ff} and F_{Fr}. The total secondary recoil resistance

$$R = R_r + F_{tr} + \mu R'_F$$

where

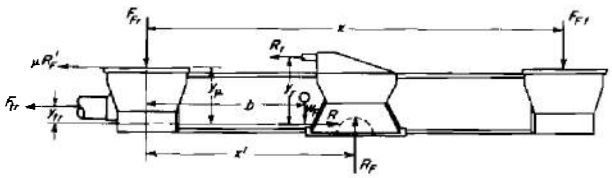
R_r = secondary recoil rod pull

R'_F = vertical load transmitted by top carriage

$\mu R'_F$ = frictional force on front bolster

* Reference 2.

F_{tr} = tie rod load



F_{Ff} = VERTICAL LOAD ON FRONT SLIDE
F_{Fr} = VERTICAL LOAD ON REAR SLIDE
F_{tr} = TIE ROD FORCE
R = TOTAL SECONDARY RECOIL FORCE
R_F = TURNTABLE BEARING VERTICAL REACTION
R_r = SECONDARY RECOIL ROD PULL
$\mu R'_F$ = FRICTIONAL FORCE ON FRONT BOLSTER
W_F = WEIGHT OF FRONT BOLSTER

FIG. 19 LOADING DIAGRAM OF FRONT BOLSTER

These components of the secondary recoil force and the vertical reaction, R'_F, are available in the top carriage analysis*. The slide loads are found by taking moments about the intersection of R and F_{Fr} and solving for

$$F_{Ff} = \frac{1}{x}\left(x'R_F - bW_F + y_rR_r + y_\mu \mu R'_F + y_{tr}F_{tr}\right) \qquad (20)$$

where

$R_F = R'_F + W_F$, vertical reaction on turntable

W_F = weight of front bolster

$R'_F = F_{Fr} + F_{Ff}$

$$F_{Fr} = R_F - W_F - F_{Ff} \qquad (21)$$

3. Distribution of Float Loads

38. During fine traverse, the float does not move while the structure above it pivots about the turntable. This activity will shift the rear support load anywhere along the span of the float but the load must be confined within the limits of fine traverse which, in effect, establishes the span or length. The width (measured along the longitudinal axis of the weapon) is selected so that the resulting ground contact area will yield pressures compatible with allowable ground contact pressures. Present practice allows an average pressure of 50 psi on spades and floats.

* Reference 2.

39. The force transmitting member or slide between bolster and float, because it travels along the float, must necessarily be shorter than the float by at least the total traversing distance. The shorter length precludes any portion at maximum traverse from extending beyond the float and becoming unsupported. A ratio of 4:1 of the lengths represents a good starting design criterion. The load distribution over the short distance is assumed to be rectangular. The reaction on the ground is found by isolating the float as a free body and then distributing these reactions for static equilibrium. Since the float is generally more rigid than the ground, consider the reaction distribution to be linear. Figure 20 represents the various loading diagrams.

Definition of Symbols

a = length of distributed load from bolster

b = length of distributed reaction on float

c = width of float

d = length (span) of float

e = distance from center of load to edge of float at 0

R_R = load on float

u = minimum unit reaction of float

w = maximum unit reaction of float

Equating the vertical forces of Figure 20(a)

$$R_R = \tfrac{1}{2}\,(u + w)b \tag{22}$$

$$w = \frac{2R_R - ub}{b} \tag{23}$$

Equating the moments about 0

$$eR_R = \frac{1}{2}\,ub^2 + \frac{1}{6}\,(w - u)\,b^2 \tag{24}$$

Substituting for w and solving for u

$$u = R_R\,\frac{6e - 2b}{b^2} \tag{25}$$

When the load is centered, $b = d$, $e = \tfrac{1}{2}d$, and $u = w$, a rectangular distribution. When R_R moves off center, the distribution becomes trapezoidal. Continued displacement to the left (Figure 20(b)) will eventually develop a triangular distribution over the full length which will happen when $e = 1/3\ d$. Further displacement of R_R still induces a triangular reaction but $b<d$ and $u = 0$. In this case, from Equation 23

$$w = \frac{2R_R}{b} \tag{26}$$

and, from Equation 24

$$eR_R = \frac{1}{6}\,wb^2 \tag{27}$$

Substituting for w in Equation 27 and solving for b

$$b = 3e \tag{28}$$

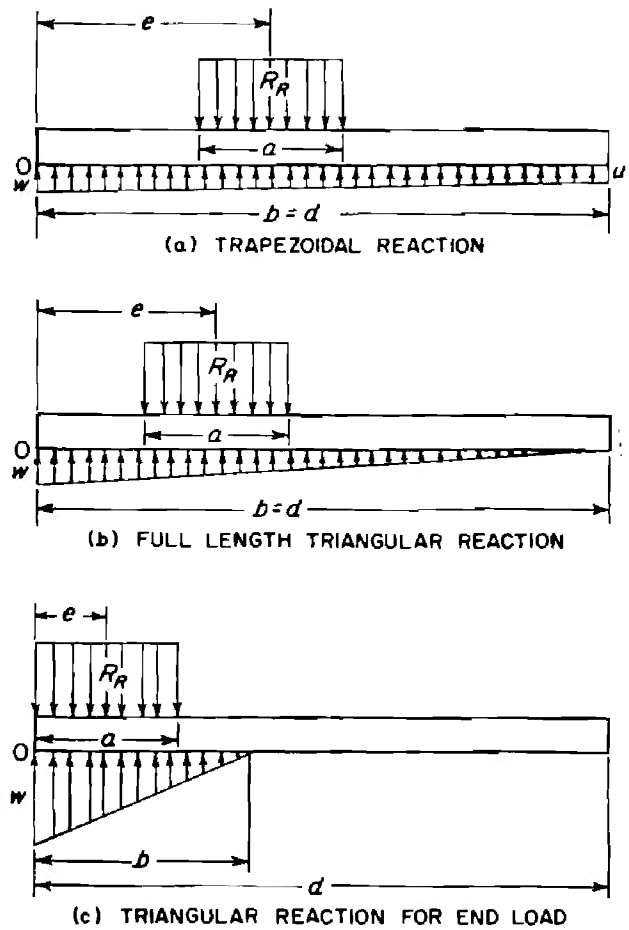


FIG. 20. LOADING DIAGRAM OF FLOAT

When R_R is on the end of the float (Figure 20(c)),

$e = \tfrac{1}{2}a$, $b = 1.5a$, and

$$w = \frac{4R_R}{3a} \tag{29}$$

This condition yields both the upper limits of maximum and average ground pressures. The average allowable pressure is 50 psi, therefore, for a triangular distribution, the maximum pressure is 100 psi

$$p_m = \frac{4}{3}\,\frac{R_R}{ac} \tag{30}$$

When $p_m = 100$ psi, the required float width

$$c = \frac{R_B}{75a} \tag{31}$$

If c becomes too large the lengths of the slide and therefore the length of the applied load, a, can be increased to reduce the required float width to a reasonable dimension. Although this last condition determines the critical ground pressure, the maximum bending moment on the float is produced by the rectangular load distribution when the applied load is centered, Figure 20(a).

4. Distribution of Turntable Loads

a. Closed Structure

40. The vertical reaction on the turntable is distributed as ground pressure to balance the couple produced by the secondary recoil force and the spade reaction. Figure 21 shows the distribution over the bottom of a closed structure. For convenience, the turntable weight is assumed to be part of the applied vertical load, R_F. The ground pressure varies from a minimum of p_1 to a maximum of p_2. These pressures are found by considering the contact area analogous to a structural member having a circular cross section and being subjected to a combined bending and axial load. When r is the turntable radius, the pressure area

$$A_t = \pi r^2 \tag{32}$$

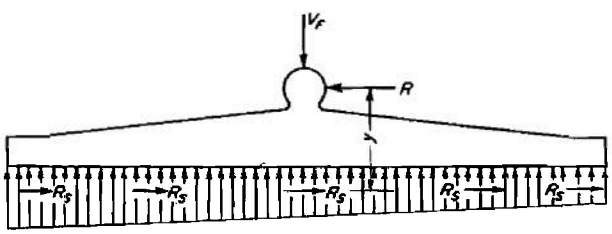


FIG.21. TURNTABLE LOADING DIAGRAM

and the equivalent section modulus is

$$Z_t = \frac{\pi}{4} r^3 \tag{33}$$

Since $\Sigma R_s = R$, the secondary recoil force, the overturning moment acting on the turntable is

$$M_t = yR \tag{34}$$

where y is the distance between secondary recoil resistance and the ground reaction on the spades. From the stress analogy, the maximum ground pressure

$$p_2 = \frac{R_F}{A_t} + \frac{M_t}{Z_t} \tag{35}$$

and the minimum ground pressure

$$p_1 = \frac{R_F}{A_t} - \frac{M_t}{Z_t} \tag{36}$$

41. The bending moment at any position due to ground pressure is found by integrating the pressure moment across the area. The pressure force is shown as a volume over half a circular segment in Figure 22. The differential moment for one element about the y-axis (diameter)

$$dM_p = 2xdV = 2xdxdyp_x \tag{37a}$$

where

$$p_x = \frac{p_1 - p_2}{2r}x + \frac{p_1 + p_2}{2} \tag{37b}$$

$$y^2 = r^2 - x^2 \qquad (37c)$$

Integrating

$$M_p = 2 \int_b^r x dx \int_0^{\sqrt{r^2 - x^2}} p_x dy \qquad (38a)$$

$$M_p = \frac{p_1 - p_2}{r} \int_b^r x^2 \sqrt{r^2 - x^2}\, dx + (p_1 + p_2)$$

$$\int_b^r x \sqrt{r^2 - x^2}\, dx \qquad (38b)$$

$$M_p = \frac{p_1 - p_2}{r} \left[\frac{r^4}{16} \left(\pi - 2 \sin^{-1} \frac{b}{r} \right) + \frac{b}{4} (r^2 - b^2)^{3/2} - \frac{r^2 b}{8} (r^2 - b^2)^{1/2} \right] + \frac{p_1 + p_2}{3} (r^2 - b^2)^{3/2} \qquad (38c)$$

The distance to the section may vary from $b = r$ to $b = -r$. The total bending moment is found by superposing the moments due to other applied loads.

b. Ring and Spoke Structure

42. The loading analysis is based on a statically indeterminate elastic structure supported on an elastic foundation (earth). Although the earth is not a true elastic medium with a uniform and constant foundation modulus, it may be treated as such when the modulus is relatively low. The loading on the turntable is unsymmetrical due to the couple imposed on it by the horizontal secondary recoil force and spade reactions.

43. Let the ring be divided into sections of equal length (Figure 23(a)), one attached to each spoke so that the bearing force on each segment can be assumed to act on its spoke. The distribution of loads and moments is determined by investigating the linear and angular deflections at the joints between spoke and ring at A and between spoke and hub at B (Figure 23(b)).

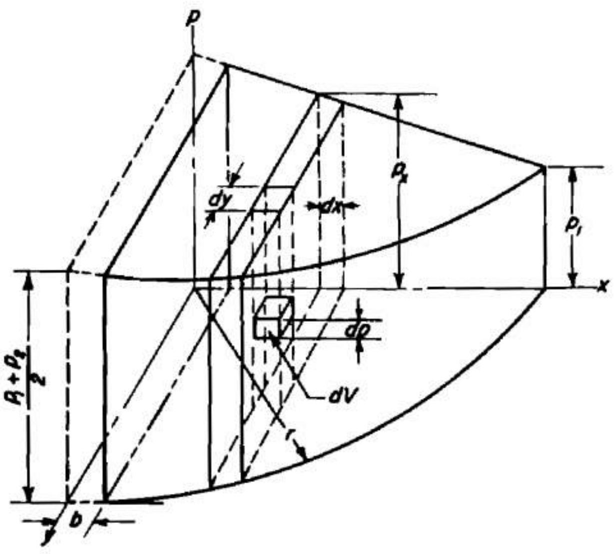


FIG. 22. GROUND PRESSURE DIAGRAM

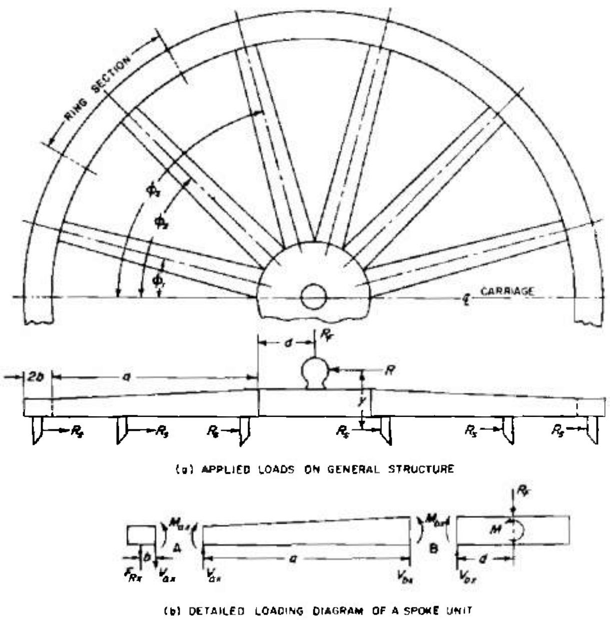


FIG 23 LOADING DIAGRAM OF RING AND SPOKE TURNTABLE

Definitions of Terms

A_h = contact area of hub

A_r = contact area of ring segment

a = length of spoke

b = half the width of ring

d = radius of hub

e = average width of spoke

F_{rx} = ring load on each segment

k_o = foundation modulus

M = applied moment at hub

M_{ax} = bending moment at A

M_{bx} = bending moment at B

n = number of spokes or segments

R_F = applied vertical load on hub

V_{ax} = vertical shear at A

V_{bx} = vertical shear at B

x = subscript designating any spoke generally

y = distance between horizontal load and spade reaction

y_{ax} = linear deflection at A, positive downward

y_{bx} = linear deflection at B, positive downward

y_h = displacement of hub

β = simplified expression in equation for beams on an elastic foundation

θ = angular deflection of hub, positive clockwise

θ_{ax} = angular deflection at A, positive clockwise

θ_{bx} = angular deflection at B, positive clockwise

λ = characteristic of the equations for beams on an elastic foundation

ϕ_x = location of spokes

The displacement of a ring section corresponding to the deflection of its spoke becomes

$$y_{rx} = y_{ax} - b\theta_{ax} \tag{39}$$

Thus, the ring load and corresponding shear at A becomes

$$V_{ax} = F_{rx} = A_r k_o y_{rx} = A_r k_o \, (y_{ax} - b\theta_{ax}) \tag{40}$$

M_{ax} is conservatively assumed to be zero since the action of the ring tends to induce a relieving moment at A. Therefore $M_{ax} = 0$

Balancing the vertical forces on the hub

$$R_F = \sum_{x=1}^{n} V_b + A_h k_o y_h \tag{41}$$

Solving for the displacement of hub

$$y_h = \frac{R_F - \sum_{x=1}^{n} V_{bx}}{A_h k_o} \qquad (42)$$

The linear deflection at the hub

$$y_{bx} = y_h - d\theta_{bx} = \frac{R_F - \sum_{x=1}^{n} V_{bx}}{A_n k_o} - d\,\theta \cos\phi_x \qquad (43)$$

Since the angular deflection at the hub

$$\theta_{bx} = \theta \cos\phi_x \qquad (44)$$

Balancing the moments at the hub

$$\sum_{x=1}^{n} dV_{bx}\cos\phi_x + \sum_{x=1}^{n} M_{bx}\cos\phi_x - M = 0 \qquad (45)$$

44. The deflections of the spokes are found by superposing the results of appropriate applications for the expressions in the equations for beams with free ends resting on an elastic foundation, types e and d*.

$$y_{ax} = -\frac{2\lambda V_{ax}}{ek_o}\,\frac{\operatorname{Sinh}\beta\operatorname{Cosh}\beta - \sin\beta\cos\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} + \frac{2\lambda V_{bx}}{ek_o}\,\frac{\operatorname{Sinh}\beta\cos\beta - \sin\beta\operatorname{Cosh}\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} + \frac{4\lambda^2 M_{bx}}{ek_o}\,\frac{\operatorname{Sinh}\beta\sin\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} \qquad (46a)$$

$$y_{bx} = -\frac{2\lambda V_{ax}}{ek_o}\,\frac{\operatorname{Sinh}\beta\cos\beta - \sin\beta\operatorname{Cosh}\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} + \frac{2\lambda V_{bx}}{ek_o}\,\frac{\operatorname{Sinh}\beta\operatorname{Cosh}\beta - \sin\beta\cos\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} - \frac{2\lambda^2 M_{bx}}{ek_o}\,\frac{\operatorname{Sinh}^2\beta + \sin^2\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} \qquad (46b)$$

$$\theta_{bx} = \frac{2\lambda^2 V_{ax}}{ek_o}\,\frac{\operatorname{Sinh}^2\beta + \sin^2\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} + \frac{4\lambda^2 V_{bx}}{ek_o}\,\frac{\operatorname{Sinh}\beta\sin\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} - \frac{4\lambda^3 M_{bx}}{ek_o}\,\frac{\operatorname{Sinh}\beta\cos\beta + \sin\beta\operatorname{Cosh}\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} \qquad (46c)$$

$$\theta_{bx} = \frac{4\lambda^2 V_{ax}}{ek_o}\,\frac{\operatorname{Sinh}\beta\sin\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} + \frac{2\lambda^2 V_{bx}}{ek_o}\,\frac{\operatorname{Sinh}^2\beta + \sin^2\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} - \frac{4\lambda^3 M_{bx}}{ek_o}\,\frac{\operatorname{Sinh}\beta\operatorname{Cosh}\beta + \sin\beta\cos\beta}{\operatorname{Sinh}^2\beta - \sin^2\beta} \qquad (46d)$$

The equations become more wieldy by assigning reasonable values to the constant terms.

$a = 21.5$ in

$b = 1.75$ in

* Reference 5, pp. 52-53.

$d = 5.0$ in

$e = 2.25$ in

$y = 10$ in

$k_o = 100$ lb/in^3

$M = Ry = 120{,}000$ lb-in

$n = 12$

$R = 12{,}000$ lb, secondary recoil force

$R_F = 18{,}850$ lb

$E = 10.3 \times 10^6$ lb/in^2, modulus of elasticity of aluminum

$\overline{R} = 28$ in, mean radius of ring

$I = 2.39$ in^4, moment of inertia of a spoke section

$$\lambda = \left[\frac{ek_o}{4EI}\right]^{1/4} = \left[\frac{2.25 \times 100}{4 \times 10.3 \times 10^6 \times 2.39}\right]^{1/4}$$

$$= \sqrt[4]{.228 \times 10^{-4}} = .0389 \text{ in}^{-1}$$

$$\beta = \lambda a = .0389 \times 21.5 = 0.837 \text{ radian} = 47^\circ\ 58'$$

$\text{Sinh}\ \beta = .938$

$\text{Cosh}\ \beta = 1.371$

$\sin \beta = .743$

$\cos \beta = .669$

$A_h k_o = \pi d^2 k_o = 7850$ lb/in

$$\frac{2\lambda}{ek_o} = 34.6 \times 10^{-5} \text{ in/lb}$$

$$\frac{2\lambda^2}{ek_o} = 1.34 \times 10^{-5} \text{ lb}^{-1}$$

$$\frac{4\lambda^2}{ek_o} = .2.68 \times 10^{-5} \text{ lb}^{-1}$$

$$\frac{4\lambda^3}{ek_o} = 0.1045 \times 10^{-5} \text{ in}^{-1} \text{ lb}^{-1}$$

$$A_r k_o = \frac{\pi}{12}\left[(\overline{R} + b)^2 - (\overline{R}\text{-}b)^2\right] k_o$$

$$= 5100 \text{ lb/in}$$

Substituting the numerical values in the deflection Equations 46a to 46d and collecting terms.

$$y_{ax} = (-83.4V_{ax} - 41.3V_{bx} + 5.7M_{bx})10^{-5} \tag{47a}$$

$$y_{bx} = (41.3V_{ax} + 83.4V_{bx} - 5.85M_{bx})10^{-5} \tag{47b}$$

$$\theta_{ax} = (5.85V_{ax} + 5.7V_{bx} - 0.525M_{bx})10^{-5} \tag{47c}$$

$$\theta_{bx} = (5.7V_{ax} + 5.85V_{bx} - 0.568M_{bx})10^{-5} \tag{47d}$$

45. With $n = 12$, the turntable may be arranged to be symmetrically loaded and only one side need be investigated, therefore only 6 equations are needed for each series. The first series of 6 equations are derived from the relationship indicated in Equation 40, from which

$$y_{rx} = \frac{F_{rx}}{A_r k_o} = \frac{F_{rx}}{5100} = 19.6 \times 10^{-5} F_{rx}$$

Equation 39 may be written

$$19.6 \times 10^{-5} F_{rx} = y_{ax} - b\theta_{ax} \tag{48}$$

Substituting the expression for y_{ax} (Equation 47a) and for θ_{ax} (Equation 47c) and noting that $V_{ax} = F_{rx}$ (Equation 40), Equation 48 may be expressed as

$$113.2\, F_{rx} + 51.3\, V_{bx} - 6.62 M_{bx} = 0 \tag{49}$$

The second series of 6 equations is derived from Equation 44 by substituting the expression for θ_{bx} (Equation 47d)

$$5.7\, F_{rx} + 5.85 V_{bx} - 0.568 M_{bx}$$
$$- \theta \cos \phi_x \times 10^6 = 0 \tag{50}$$

The third series of 6 equations is derived from Equation 43. Since $n = 12$ and the turntable is symmetrical, $\sum_{x=1}^{n}$ may be written $2\sum_{x=1}^{6}$. Substituting for y_{bx}

(Equation 47b) and for $\theta \cos \phi_x$ (Equation 50)

Equation 43 becomes

$$(41.3 F_{rx} + 83.4 V_{bx} - 5.85 M_{bx}) 10^{-5}$$

$$= \frac{R_F}{A_n k_o} - \frac{2\sum_{x=1}^{6} V_b}{A_n k_o}$$

$$-5\,(5.7 F_{rx} + 5.85 V_{bx} - 0.568 M_{bx}) 10^{-5} \tag{51a}$$

but

$$\sum_{x=1}^{6} V_{bx} = V_{b1} + V_{b2} + V_{b3} + V_{b4}$$

$+ V_{b5} + V_{b6}$, $R_F = 18{,}850$ lb. and

$A_n k_o = 7850$ lb/in reducing the equation to

$$69.8 F_{rx} + 25.4\,(V_{b1} + V_{b2} + V_{b3} + V_{b4}$$
$$+ V_{b5} + V_{b6}) + 112.6 V_{bx} - 8.69 M_{bx}$$
$$- 240{,}100 = 0 \tag{51b}$$

The 19th equation is obtained by expanding Equation 45. When $M = 120{,}000$ lb-in. and $d = 5.0$ in.

$$10(V_{b1} \cos \phi_1 + \cdots + V_{b6} \cos \phi_6)$$
$$+ 2(M_{b1} \cos \phi_1 + \cdots M_{b6} \cos \phi_6)$$
$$-120{,}000 = 0 \tag{52}$$

Values of ϕ_x and $\cos \phi_x$

$\phi_1 = 15°$	$\cos \phi_1 = 0.966$
$\phi_2 = 45°$	$\cos \phi_2 = 0.707$
$\phi_3 = 75°$	$\cos \phi_3 = 0.259$

$\phi_4 = 105^\circ$ $\quad\quad \cos\phi_4 = -0.259$

$\phi_5 = 135^\circ$ $\quad\quad \cos\phi_5 = -0.707$

$\phi_6 = 165^\circ$ $\quad\quad \cos\phi_6 = -0.966$

The 19 equations in final form are

$$113.2\ F_{r1} + 51.3V_{b1} - 6.62M_{b1} = 0 \tag{53a}$$

$$113.2\ F_{r2} + 51.3V_{b2} - 6.62M_{b2} = 0 \tag{53b}$$

$$113.2\ F_{r3} + 51.3V_{b3} - 6.62M_{b3} = 0 \tag{53c}$$

$$113.2\ F_{r4} + 51.3V_{b4} - 6.62M_{b4} = 0 \tag{53d}$$

$$113.2\ F_{r5} + 51.3V_{b5} - 6.62M_{b5} = 0 \tag{53e}$$

$$113.2\ F_{r6} + 51.3V_{b6} - 6.62M_{b6} = 0 \tag{53f}$$

$$5.7\ F_{r1} + 5.85V_{b1} - .568M_{b1} - 96600\ \theta = 0 \tag{53g}$$

$$5.7\ F_{r2} + 5.85V_{b2} - .568M_{62} - 70700\ \theta = 0 \tag{53h}$$

$$5.7\ F_{r3} + 5.85V_{b3} - .568M_{b3} - 25900\ \theta = 0 \tag{53i}$$

$$5.7\ F_{r4} + 5.85V_{b4} - .568M_{b4} - 25900\ \theta = 0 \tag{53j}$$

$$5.7\ F_{r5} + 5.85V_{b5} - .568M_{b5} - 70700\ \theta = 0 \tag{53k}$$

$$5.7\ F_{r6} + 5.85V_{b6} - .568M_{b6} - 96600\ \theta = 0 \tag{53l}$$

$$69.8\ F_{r1} + 138V_{b1} + 25.4(V_{b2} + V_{b3} + V_{b4} + V_{b5} + V_{b6}) - 8.69M_{b1} - 240{,}100 = 0 \tag{53m}$$

$$69.8\ F_{r2} + 138V_{b2} + 25.4(V_{b1} + V_{b3} + V_{b4} + V_{b5} + V_{b6}) - 8.69M_{b2} - 240{,}100 = 0 \tag{53n}$$

$$69.8\ F_{r3} + 138V_{b3} + 25.4(V_{b1} + V_{b2} + V_{b4} + V_{b5} + V_{b6}) - 8.69M_{b3} - 240{,}100 = 0 \tag{53o}$$

$$69.8\ F_{r4} + 138V_{b4} + 25.4(V_{b1} + V_{b2} + V_{b3} + V_{b5} + V_{b6}) - 8.69M_{b4} - 240{,}100 = 0 \tag{53p}$$

$$69.8\ F_{r5} + 138V_{b5} + 25.4(V_{b1} + V_{b2} + V_{b3} + V_{b4} + V_{b6}) - 8.69M_{b5} - 240{,}100 = 0 \tag{53q}$$

$$69.8\ F_{r6} + 138V_{b6} + 25.4(V_{h1} + V_{b2} + V_{b3} + V_{b4} + V_{b5}) - 8.69M_{b6} - 240{,}100 = 0 \tag{53r}$$

$$\begin{aligned} &9.66V_{b1} + 7.07V_{b2} + 2.59V_{b3} - 2.59V_{b4} \\ &- 7.07V_{b5} - 9.66V_{b6} + 1.932M_{b1} + 1.414M_{b2} \\ &+ .518M_{b3} - .518M_{b4} - 1.414M_{b5} \\ &- 1.932M_{b6} - 120{,}000 = 0 \end{aligned} \tag{53s}$$

The nineteen unknown quantities are readily determined with an electronic computer.

$F_{r1} = 1004$ lb $\quad V_{b1} = 2275$ lb

$M_{b1} = 34{,}800$ lb-in

$F_{r2} = 872$ lb $\quad V_{b2} = 2045$ lb

$M_{b2} = 30{,}800$ lb-in

$F_{r3} = 644$ lb $\quad V_{b3} = 1648$ lb

$M_{b3} = 23{,}800$ lb-in

$F_{r4} = 380$ lb $\quad V_{b4} = 1189$ lb

$M_{b4} = 15{,}720$ lb-in

$F_{r5} = 152$ lb $\quad V_{b5} = 792$ lb

$M_{b5} = 8{,}750$ lb-in

$F_{r6} = 21$ lb $\quad V_{b6} = 563$ lb

$M_{b6} = 4{,}710$ lb-in

$$\theta = -0.00759 \text{ radian}$$

The maximum bending stress on the spokes

$$\sigma = \frac{B_{b1}c}{I} = 23{,}700 \text{ lb/in}^2$$

where c = 1.625 in, (total depth of section is 3.25 in)

$$I = 2.39 \text{ in}^4$$

The material is aluminum alloy with a yield strength of 44,000 lb/in².

The factor of safety

$$S_f = \frac{40{,}000}{23{,}700} = 1.68$$

V. SAMPLE CALCULATIONS

A. SINGLE RECOIL—SPLIT TRAIL TYPE

46. The following design data are assigned to a bottom carriage similar to the type shown in Figure 13. The weight of the complete structure is assumed to be incorporated in the applied loads which have the same values as the reactions on the top carriage.*

$a = 11.75$ in	$x_b = 12$ in
$d = 14$ in	$y_t = 10$ in
$x_a = 4.7$ in	$\epsilon = 45^\circ$
$y_a = 14$ in	$F_b = 297{,}000$ lb
$\beta = 20^\circ$	$c = 17$ in
$F_a = 437{,}000$ lb	$L = 180$ in
$b = 6$ in	$L_p = 19$ in
$h = 15.5$ in	$F_h = 6{,}800$ lb
	$\psi = 5^\circ$

On the basis of Equation 4,

$$R_{s1} = F_h \frac{\cos\beta + \sin\beta \cot\epsilon}{2 \cos\epsilon}$$

$$= 68{,}600 \frac{.940 + .342 \times 1.0}{2 \times .707} = 62{,}200 \text{ lb}$$

Similarly, from Equation 3,

$$R_{s2} = R_{s1} - F_h \frac{\sin\beta}{\sin\epsilon}$$

$$= 62{,}200 - 68{,}600 \frac{.342}{.707} = 29{,}000 \text{ lb}$$

In Figure 13, take vertical moments about an axis parallel to the x-axis where R_j and the x-y plane intersect.

$$(R_{f1} + R_{f2}) \left[(b + L\cos\psi) \cos\epsilon + y_a + y_t\right]$$
$$= (x_aF_a + x_bF_b + dF_h) \cos\beta + y_a (F_a - F_b)$$
$$+ (a+h) (R_s + R_s) \cos\epsilon$$

$$(R_{f1} + R_{f2}) \left[(6+180\times.996) .707+24\right]$$
$$=(4.7\times437{,}000+12\times297{,}000+14\times68{,}600) .94$$
$$+ 14\times140{,}000 + 27.25\times91{,}200\times.707$$

$$155(R_{f1} + R_{f2}) = 6{,}190{,}000 + 1{,}960{,}000$$
$$+ 1{,}760{,}000 = 9{,}910{,}000 \text{ lb-in}$$

$$R_{f1} + R_{f2} = 63{,}900$$

Again in Figure 13, take vertical moments about the y-axis

* Reference 2, page 19, paragraph 61.

$$(R_{f1} - R_{f2})\ [(b+L\cos\psi)\sin\epsilon + c] =$$

$$(x_a F_a + x_b F_b + dF_h)\sin\beta + (a+h)\ (R_s - R_s)\sin\epsilon$$

$$(R_{f1} - R_{f2})\ [(6+180\times.996)\ .707+17]$$

$$=(4.7\times437{,}000+12\times297{,}000+14\times68{,}600)\ .342$$

$$+27.5 \times 33{,}200 \times .707$$

$$148\ (R_{f1} - R_{f2}) = 2{,}250{,}000 + 640{,}000$$

$$= 2{,}890{,}000 \text{ lb-in}$$

$$R_{f1} - R_{f2} = 19{,}500 \text{ lb}$$

$$R_{f1} = 41{,}700 \text{ lb}$$

$$R_{f2} = 22{,}200 \text{ lb}$$

1. Trail Stresses

47. The analysis continues for the left trail which carries the larger loads. From Equation 8

$$F_c = R_{f1} \sin\psi + R_{s1}\cos\psi = 41{,}700\times.0872$$

$$+ 62{,}200\times.996 = 65{,}600 \text{ lb}$$

Equation 9 shows that

$$F_n = R_{f1}\cos\psi - R_{s1}\sin\psi = 41{,}700\times.996$$

$$- 62{,}200\times.0872 = 36{,}200 \text{ lb}$$

The applied moment according to Equation 10

$$M_s = bR_{f1} - aR_{s1} = 6 \times 41{,}700$$

$$- 11.75 \times 62{,}200 = -481{,}000 \text{ lb-in}$$

The bending moment at the section adjacent to the pin is

$$M = M_s + LF_n = -481{,}000 + 180 \times 36{,}200$$

$$= 6{,}039{,}000 \text{ lb-in}$$

Although an additional moment will be induced by the axial force about the eccentricity created by the deflection of the beam, it will be negligible in comparison with the other applied moments.

The trail is a welded box construction of 3/8-in. plates and measures 10.5 in. wide and 21 in. deep at the section.

$$I = \frac{1}{12}\ (10.5\times21^3 - 9.75\times20.25^3)$$

$$= \frac{1}{12}\ (97{,}200 - 81{,}200) = 1333 \text{ in}^4$$

$$\sigma_b = \frac{Mc}{I} = \frac{6{,}039{,}000 \times 10.5}{1333}$$

$$= 47{,}500 \text{ lb/in}^2$$

The cross sectional area is

$$A = 10.5 \times 21 - 9.75 \times 20.25 = 23 \text{ in}^2$$

$$\sigma_c = \frac{F_c}{A} = \frac{65600}{23} = 2{,}900 \text{ lb/in}^2$$

$$\sigma = \sigma_b + \sigma_c = 50{,}400 \text{ lb/in}^2$$

The yield strength of the material, $\sigma_y = 90{,}000$ lb/in^2 (assumed). The factor of safety

$$S_f = \frac{\sigma_y}{\sigma_c} = 1.79$$

The maximum shear stress on the weld seam of the box construction occurs at the spade end of the trail where the section is 10.5 in $\times$ 8.5 in. deep.

$$I = \frac{1}{12} (10.5 \times 8.5^3 - 9.75 \times 7.75^3)$$

$$= \frac{1}{12} (6440 - 4540) = 158.2 \text{ in}^4$$

The area of the top plate, $A_p = 10.5 \times .375$

$$= 3.94 \text{ in}^2$$

The shear stress on the weld

$$\tau = \frac{F_n A_c \bar{x}}{2tI} = \frac{36{,}200 \times 3.94 \times 4.07}{2 \times .375 \times 158.2}$$

$$= 4{,}900 \text{ lb/in}^2 \text{ (not significant)}$$

2. Trail Pin

48. The reactions on the trail pin are resolved into perpendicular and axial forces (Figure 24). The component of the trail forces parallel to the pin pulls the trail off the lower pin lug of the bottom carriage, thereby applying all this force to the upper lug. From Equation 11, the top reaction on the hinge pin

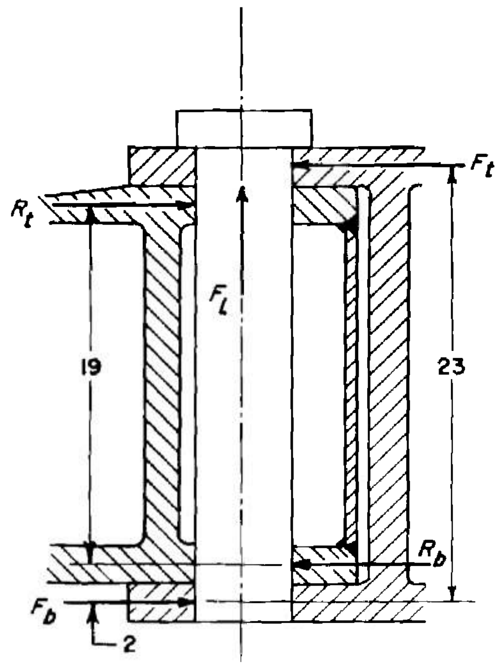


FIG. 24. TRAIL PIN LOAD

$$R_t = \frac{M_s + LF_n}{L_p} + \tfrac{1}{2}R_{s1} =$$

$$= \frac{-481{,}000 + 180 \times 36{,}200}{19}$$

$$+ 31{,}100 = 349{,}100 \text{ lb}$$

where L_p is the center to center distance of the trail pin lugs.

Equation 12 has

$$R_b = R_t - R_{s1} = 349{,}100 - 62{,}200 = 286{,}900 \text{ lb}$$

The vertical load on the upper lug is the transmitted float load

$$F_L = R_{f1} = 41{,}700 \text{ lb}$$

The pin is treated as a beam with the trail lugs providing the applied loads and the bottom carriage lugs, the outer supports. Taking moments about F_b (Figure 24)

$$23F_t = 21R_t - 2R_b = 6{,}751{,}000 \text{ lb-in}$$

$$F_t = 293{,}500 \text{ lb}$$

$$F_b = F_t - R_s = 231{,}300 \text{ lb}$$

The pin is 5 in OD and 3¼ in ID

$$I = \frac{\pi}{64}(5.0^4 - 3.25^4) = 25.2 \text{ in}^4$$

The maximum bending moment is at the center of the upper trail lug

$$M = 2F_t = 587{,}000 \text{ lb-in}$$

$$\sigma = \frac{Mc}{I} = \frac{587{,}000 \times 2.5}{25.2}$$

$$= 58{,}300 \text{ lb/in}^2$$

$$S_f = \frac{\sigma_y}{\sigma} = \frac{90{,}000}{58{,}300} = 1.54$$

The shear area of this pin

$$A = \frac{\pi}{4}(5.0^2 - 3.25^2) = 11.35 \text{ in}^2$$

$$\tau = \frac{F_t}{A} = \frac{293{,}500}{11.35} = 25{,}900 \text{ lb/in}^2$$

$$S_f = \frac{0.6\sigma_y}{\tau} = \frac{54{,}000}{25{,}900} = 2.08$$

The value $0.6\sigma_y$ is generally considered to be the yield strength of steel in shear.

3. Lug Stresses

49. The direction of the applied load, F_L, subjects the lower lug on the bottom carriage to a combined bending and tensile load. Here the lug is assumed to be a curved beam, fixed at the ends and carrying a uniform load (Figure 25). A procedure similar to that for a concentrated internal load on a ring* shows that for a uniformly distributed load, the maximum bending moment for either end is

$$M = -\frac{1}{8}RF_b = -\frac{1}{8} \times 4.25 \times 231{,}300$$

$$-123{,}000 \text{ lb-in}$$

Figure 25 shows the pertinent dimensions.

$b = 2.0$ in

$h = 3.5$ in

$c = 1.75$ in

$y = -1.75$ in, distance from centroidal axis to concave side

$R = 4.25$ in, radius of centroidal axis

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* Reference 6, Article 56.

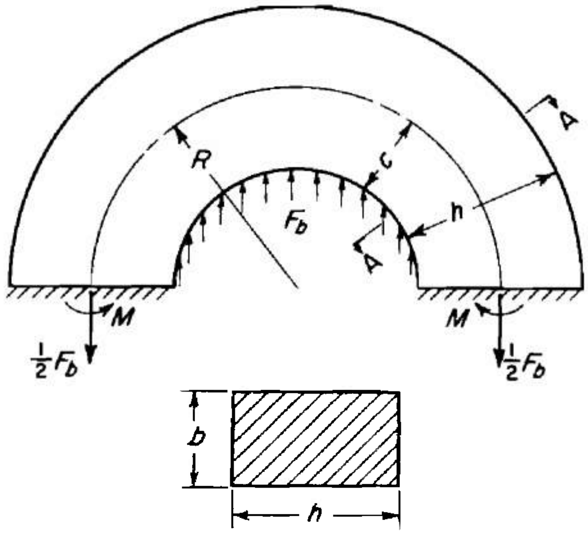

FIG. 25. LOADING DIAGRAM OF LUG

The curved beam factor

$$Z = \frac{1}{3}\left(\frac{c}{R}\right)^2 \times \frac{1}{5}\left(\frac{c}{R}\right)^4$$

$$+ \frac{1}{7}\left(\frac{c}{r}\right)^6 = .0631^*$$

The cross sectional area

$$A = bh = 7.0 \text{ in}^2$$

The combined bending and tensile stress

$$\sigma = \frac{F_b}{2A} + \frac{M}{AR}\left(1 + \frac{1}{Z}\frac{y}{R+y}\right)^\dagger$$

$$\sigma = \frac{231300}{14} + \frac{-124000}{7 \times 4.25}\left(1 + 15.85\frac{-1.75}{2.5}\right)$$

$$= 58{,}600 \text{ lb/in}^2$$

* Reference 6, Equation 763, Page 663.
† Reference 6, Equation 204, Page 180.

$$S_f = \frac{\sigma_y}{\sigma} = \frac{90{,}000}{58{,}600} = 1.54$$

The direct shear stress through the ring

$$\tau = \frac{F_b}{2A} = \frac{231{,}300}{14} = 16{,}500 \text{ lb/in}^2$$

$$S_f = \frac{0.6\ \sigma_y}{\tau} = \frac{0.6 \times 90{,}000}{16{,}500} = 3.27$$

B. DOUBLE RECOIL TYPE—STABILITY

50. A number of conditions exist for determining the stability during the recoil cycle. During recoil, the stability is determined with the help of Figure 15 and by Equation 13. The assumed dimensions and weights are compatible with structures of this type. The applied loads are the same as those in the cradle analysis.‡

1. Dimensions (inches)

$x_1 = 98$

$x_2 = 119$

$x_T = 100$

$x_t = x_T + r_t \cos\theta$

$r_t = 60.75$

$y_g = x_T \sin\theta - y_T \cos\theta$

$y_T = 60$

$y_a = y_g - 0.10$

$y_2 = 34.5$

$y_t = y_T + r_t \sin\theta$

$y_s = 12$

‡ Reference 7, Page 23, Paragraph 61.

2. Forces (pounds)

$W_{BC} = 2{,}000$, weight of bottom carriage

$W_{TC} = 10{,}000$, weight of top carriage

$W_t = 14{,}000$, weight of tipping parts

$K = 150{,}000$, total resistance to primary recoil

$R = 40{,}000$, total resistance to secondary recoil

$F_g = 1{,}810{,}000$, maximum propellant gas force

$F_a = 1{,}660{,}000$, maximum inertia force of primary parts

The longitudinal ground reaction, i.e., spade load, is equal to the total resistance to secondary recoil, $R_s = R$. After the secondary recoil acceleration is found*, the inertia forces of the top carriage and tipping parts due to secondary acceleration may be expressed as

$$F_{TC} = W_{TC}a_2 \qquad (54)$$

$$F_t = W_t a_2 \qquad (55)$$

where a_2 = secondary recoil acceleration in terms of gravity.

Solve for F_t by dividing Equation 55 by Equation 54.

When numerical values are substituted for W_t and W_{TC}, we find that

$$F_t = 1.4\, F_{TC}$$

Balancing the horizontal components of the forces in Figure 15.

$$F_{TC} + F_t = (F_g - F_a)\cos\theta - R_s$$

* **Reference 4, Equation 88.**

$$F_{TC} = \frac{(F_g - F_a)\cos\theta - R_s}{2.4}$$

All loading conditions during the recoil cycle should be investigated. Three are shown here. Two are solutions for the minimum angle of elevation, the other involves the maximum angle for stability.

When the maximum propellant gas force is acting, secondary recoil forces have not appeared, i.e., F_t and F_{TC} are zero. All reactions are in the form of inertia forces. For this condition, the minimum angle of elevation is found by equating Equation 13 to zero. Substituting numerical values, the various components of Equation 13 become, in lb-in.

$$y_gF_g - y_aF_a = 150{,}000y_g + 166{,}000$$
$$= (15\sin\theta - 9\cos\theta)\,10^6 + 166{,}000$$

$$x_1W_{BC} = 98 \times 2000 = 196{,}000$$

$$x_2W_{TC} = 118.9 \times 10{,}000 = 1{,}189{,}000$$

$$y_2F_{TC} = 34.5 \times 150{,}000\cos\theta/2.4 = 2{,}156{,}000\cos\theta$$

$$x_tW_t = (100 + 60.75\cos\theta)\,14{,}000$$
$$= 1{,}400{,}000 + 850{,}000\cos\theta$$

$$y_tF_t = (60 + 60.75\sin\theta)\,87{,}500\cos\theta$$
$$= (5.25\cos\theta + 5.32\sin\theta\cos\theta)\,10^6$$

$$y_sR_s = 12 \times 0 = 0$$

Collecting terms and reducing to lower constants, Equation 13 becomes

$$150\sin\theta - 7.44\cos\theta + 53.2\sin\theta\cos\theta$$
$$+ 29.51 = 0$$

Which shows by iterative substitution of the trigonometric functions that the gun is stable in primary recoil for an angle of elevation of less than —5°.

After propellant gas forces cease, and the secondary recoil becomes active, $F_a = -K = -150{,}000$ lb., and $R_s = 40{,}000$ lb

$$F_g y_g - F_a y_a = 0 + 150{,}000 y_g - 15{,}000$$
$$= (15 \sin\theta - 9 \cos\theta)10^6 - 15{,}000$$

$$y_2 F_{TC} = 34.5\ (62{,}500 \cos\theta - 16{,}700)$$
$$= 2{,}156{,}000 \cos\theta - 576{,}000$$

$$y_t F_t = (60 + 60.75 \sin\theta)\ (87{,}500 \cos\theta - 23{,}400)$$
$$= (5.25 \cos\theta - 1.404 + 5.32 \sin\theta \cos\theta$$
$$- 1.421 \sin\theta)\ 10^6$$

$$-y_s R_s = -12 \times 40{,}000 = -480{,}000$$

Other components are the same as in the previous case. After substitution and reduction, Equation 13 becomes

$$135.8 \sin\theta - 7.44 \cos\theta + 53.2 \sin\theta \cos\theta$$
$$+ 3.10 = 0$$

Which shows a minimum angle of elevation of 1°19' for stability. Therefore, the controlling factor for minimum angle of elevation is secondary recoil.

51. The maximum angle of elevation is determined from the condition when only secondary recoil activity is in progress.

$$F_g = F_a = K = 0$$

$$R_s = 40{,}000 \text{ lb.}$$

$$F_{TC} = \frac{-R_s}{2.4} = \frac{-40{,}000}{2.4} = -16{,}700 \text{ lb}$$

$$F_t = 1.4\ F_{TC} = -23{,}300$$

The F_g and F_a components of Equation 13 are zero, the other components are the same as for the previous case except

$$y_2 F_{TC} = 34.5 F_{TC} = -576{,}000$$

$$y_t F_t = (60 + 60.75 \sin\theta) F_t = -(1.398$$
$$+ 1.415 \sin\theta)\ 10^6$$

Substituting and reducing terms in Equation 13

$$8.5 \cos\theta - 14.15 \sin\theta + 3.31 = 0$$

Solving for θ shows a maximum angle of elevation 42°33'. Other conditions are similarly investigated. Equation 14 determines stability during buffing. Data and calculations are not given since computation procedures would be repetitive.

C. FLOAT SIZE

52. Determine the float size for the weapon of the preceding problem. The maximum load will occur after the propellant gases are no longer effective and when secondary recoil forces become active at an angle of elevation of = 42° 33'. According to the second problem of paragraph 50, Equation 13 may be written

$$xR_F = (135.8 \sin\theta - 7.44 \cos\theta$$
$$+ 53.2 \sin\theta \cos\theta + 3.1)10^5 = 11{,}570{,}000 \text{ lb-in}$$

where

$x = 140$ in

$\sin\theta = .676$

$\cos\theta = .737$

The load on the turntable

$$R_F = \frac{11,570,000}{140} = 82,600 \text{ lb},$$

The load on the float

$$R_R = K \sin\theta + (W_{BC} + W_{TC} + W_t) - R_F$$

$$= 101,400 + 26,000 - 82,600 = 44,800 \text{ lb}$$

From Equation 31

$$ac = \frac{R_R}{75} = 600 \text{ in}^2$$

If the width of the float is double the length of the distributed applied load i.e., c = 2a, then a = 17.5 in. (approximately) and c = 35 in., the width of the float. According to Paragraph 39, the float length

$$d = 4a = 70 \text{ in}$$

This length to width ratio is not a rigid requirement, and may be adjusted to suit the overall dimensions of the top and bottom carriages.

The maximum bending moment on the float is applied when the ground reaction is distributed uniformly and occurs at mid span.

$$M = \frac{1}{8} R_R \text{ (d-a)} = \frac{1}{8} \times 44,800 \times 52.5$$

$$= 293,000 \text{ lb-in}$$

On the basis of direct bending and an allowable stress of 40,000 lb/in², the required section modulus

$$Z = \frac{M}{\sigma} = \frac{293,000}{40,000} = 7.325 \text{ in}^3$$

The float is a welded steel box structure (Figure 26) consisting of thin plates and reinforced with ribs of the same material in the box cavity. A depth of 3.5 in. and a plate thickness of 1/16 in. meets the strength requirements. At Section A-A of Figure 26,

$$I = \frac{1}{12} (35 \times 3.5^3 - 34.875 \times 3.375^3) = 13.33 \text{ in}^4$$

$$\sigma = \frac{Mc}{I} = \frac{293,000 \times 1.75}{13.33} = 38,400 \text{ lb/in}^2$$

D. Turntable—Closed Structure

53. The same external loads are applied to this structure as were used in derivation of the loading analysis for the ring and spoke structure, Part IVC—4b. The overall dimensions are also retained for comparison. In Figure 21

$$V_F = R_F = 18,850 \text{ lb}$$

$$\Sigma R_s = R = 12,000 \text{ lb}$$

$$y = 10 \text{ in}$$

$$r = 30 \text{ in}$$

From Equation 32

$$A_t = \pi r^2 = 2820 \text{ in}^2$$

From Equation 33

$$Z_t = \frac{\pi}{4} r^3 = 21,400 \text{ in}^3$$

From Equation 34

$$M_t = 10 \times 12,000 = 120,000 \text{ lb-in}$$

The maximum ground pressure, Equation 35, and minimum, Equation 36

$$p_2 = \frac{R_F}{A_t} + \frac{M_t}{Z_t} = 6.57 + 5.61 = 12.18 \text{ psi}$$

$$p_1 = \frac{R_F}{A_t} - \frac{M_t}{Z_t} = 6.57 - 5.61 = 0.96 \text{ psi}$$

The bending moment at the diameter due to the ground pressure is found by Equation 38f when $b = 0$.

$$M_p = r^3 \left(\frac{\pi(p_1 - p_2)}{16} + \frac{p_1 + p_2}{3} \right)$$

$$= 27{,}000 \left(-\frac{11.22\pi}{16} + \frac{13.14}{3} \right) = 59{,}000 \text{ lb-in}$$

Assuming that half the spade reactions are applied to each half of the turntable and that the neutral axis of the cross section at the diameter is at $\frac{1}{2}y$, the bending moment produced by the horizontal forces is

$$M_h = \tfrac{1}{2}y \left(R + \tfrac{1}{2}\Sigma R_s\right) = 90{,}000 \text{ lb-in}$$

The total bending moment

$$M = M_p + M_h = 149{,}000 \text{ lb-in}$$

With the same depth of section as that for the ring and spoke type but with 1/16 in. thick cover plates.

$$I = \frac{1}{12} \times 60 \; (3.25^3 - 3.125^3) = 18.5 \text{ in}^4$$

The bending stress at this section is

$$\sigma = \frac{Mc}{I} = \frac{149{,}000 \times 1.625}{18.5} = 13{,}100 \text{ lb/in}^2$$

This stress is far less than the 23,700 lb/in² for the equivalent ring and spoke structure (Paragraph 45), indicating that the open structure is the more efficient design.

54. The loads on the remaining bottom carriage structure components are determined from Equations 15 to 21. The structures are generally simple enough that stresses may be computed by ordinary analyses of bending and direct stresses.

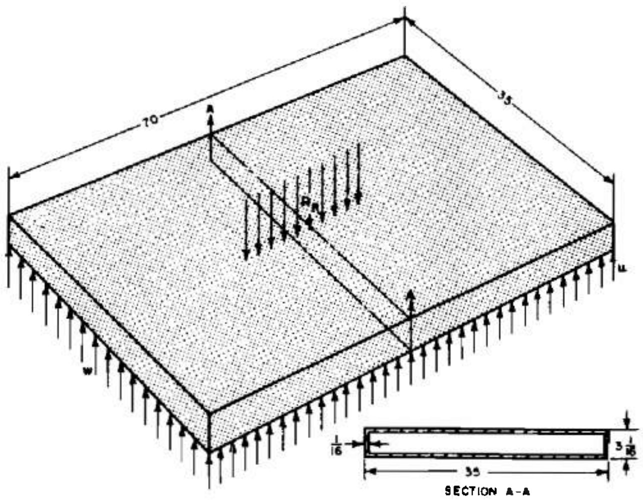

FIG 26 FLOAT WITH DISTRIBUTED LOAD AND REACTION

GLOSSARY

bogie. The rear transporting unit of a gun carriage.

bolster. On double recoil type carriages, the structure of the bottom carriage on which the top carriage rests.

bracket, supporting. Compact structure which serves as the bottom carriage on some light field pieces.

buffing. The activity of a buffer absorption of energy of counter recoil at end of stroke.

carriage. Supporting structure of a gun.

carriage, bottom. Lower supporting structure of a gun; it supports the top carriage and provides for traversing the weapon.

carriage, top. Upper supporting structure of a weapon. It supports the tipping parts and moves with the cradle in traverse.

counterrecoil. Return of a gun to the firing position after recoil.

elevation, angle of. Used in this volume for the quadrant angle of departure, the angle in the vertical plane between the horizontal and the axis of the gun bore.

emplacement. Act of fixing a gun in a prepared position from which it may be fired.

float. Horizontal structure attached to the trail or bottom carriage provided to distribute the vertical forces to the ground.

in battery. Position of gun tube fully returned from recoil.

jack. Adjustable lifting mechanism used to support and stabilize a gun carriage.

leveling socket. Structure used for supporting the top carriage of pedestal mounted guns and to provide means for leveling the top carriage.

leveling screw. Mechanism for tilting the leveling socket to bring the top carriage into a level position.

limber. Detachable or retractable unit provided with wheels, having as its function the support during transport of a gun carriage.

mount. Structure which supports a gun.

outriggers. Supporting and stabilizing structures extending outward from and attached to the bottom carriage.

pedestal. Base or support of a mount. It may serve as either top or bottom carriage depending on details of construction.

pintle. The vertical pin about which a weapon traverses.

platform, firing. Structure provided for mobile weapons as a bottom support to distrihute firing forces to the ground and to give stability.

propellant gas. Gas generated hy burning propellants.

propellant gas force. Force induced by propellant gas pressure.

recoil. Movement of the gun tube and attached parts in direction opposite to the projectile travel.

recoil system, double. System in which the gun recoils on the top carriage and the top carriage recoils on the bottom carriage.

recoil cycle. Complete sequence of recoil activity: in battery, recoil, counterrecoil, buffing, in battery.

recoil, primary. In a double recoil system, the recoil activity of primary recoiling parts.

recoil, secondary. In a double recoil system, the recoil activity of the secondary recoiling parts.

recoil system, single. System that has only the gun tube and its components as recoiling parts.

recoil force. Resistance provided to the recoiling parts by the recoil system.

GLOSSARY (continued)

recoil force, primary. Resistance provided to the movement of the primary recoiling parts by the primary recoil system.

recoil force, secondary. Resistance provided to the movement of the secondary recoiling parts by the secondary recoil system.

recoiling parts, primary. In a double recoil system, the gun tube and associated components which recoil as a unit, moving relative to the top carriage.

recoiling parts, secondary. In a double recoil system, the top carriage and components, but not including the primary recoiling parts, which move relative to the bottom carriage during recoil.

recoil mechanism, primary. The recoil mechanism provided in a double recoil system to absorb the recoil energy of the primary recoiling parts.

recoil mechanism, secondary. The recoil mechanism provided in a double recoil system to absorb the recoil energy of the secondary recoiling parts.

spade. Vertical or inclined structure to trail end or bottom carriage.

support, equalizing. The cross-beam support of an axle-support-type equalizer, on which is mounted the trails and top carriage.

trail. Rearward thrust member of a weapon. It stabilizes the weapon during firing and usually serves as a link between weapon and prime mover during transport.

traverse. Horizontal angular movement of a weapon in either direction.

traverse, coarse. General positioning of a weapon direction in azimuth.

traverse, fine. Precise positioning of a weapon direction in azimuth.

traversing axis. Axis about which the weapon rotates to change direction in azimuth.

traversing bearing. Bearing on which the traversing parts rotate.

traversing mechanism. Mechanism by which a weapon can be rotated in a horizontal plane.

traversing parts. Unit consisting of all components of a weapon that move in traverse.

turntable. Forward ground support of a double recoil type carriage on which the entire weapon may be rotated.

weapon, fixed emplacement. Weapon having a permanent emplacement.

weapon, mobile. Weapon, readily transportable and whose emplacement is usually temporary.

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